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OF

APPLIED MECHANICS

AND

MECHANICAL ENGINEERING.

IN FIVE VOLUMES.

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of Education, and Honours Certificates of the City
and Guilds of London Institute, in Mechanical
Engineering, and for Engineers Generally.

BY
ANDREW JAMIESON, M.INST.C.E.,
FORMERLY PROFESSOR OF ENGINEERING IN THE GLASGOW AND WEST OF SCOTLAND
TECHNICAL COLLEGE; MEMBER OF THE INSTITUTION OF ELECTRICAL ENGINEERS;
FELLOW OF THE ROYAL SOCIETY, EDINBURGH; AUTHOR OF TEXT-BOOKS
ON STEAM AND STEAM ENGINES, APPLIED MECHANICS AND
MECHANICAL ENGINEERING, MAGNETISM AND
ELECTRICITY, ELECTRICAL RULES
AND TABLES, ETC.

VOLUME IV.—HYDRAULICS, HYDRAULIC AND REFRIGERATING MACHINERY, WITH PNEUMATIC TOOLS.

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With Numerous Diagrams, Special Plates, and
Examination Questions.

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PREFACE

TO THE SEVENTH EDITION.

It has been found necessary to still further subdivide this wide and all-important subject of Advanced Applied Mechanics and Mechanical Engineering.

In order to do so with the least departure and derangement of the previous volumes and editions, it has been advisable and convenient to follow the recent subdivision of this subject as stated in the "Rules and Syllabus of Examinations applying to the Election of Associate Members of The Institution of Civil Engineers."

Moreover, this particular method of subdivision is practised by several Universities and Technical Colleges. It is also being advocated by Teachers in connection with the Boards of Education, and, to a certain extent, by those connected with the City and Guilds of London Examinations in Mechanical Engineering.

Consequently, Volume I. will deal with "Applied Mechanics" proper, Volume II. will discuss and give practical illustrations of "Strength and Elasticity of Materials," Volume III. will be confined to "The Theory of Structures," Volume IV. to "Hydraulics, Hydraulic and Refrigerating Machinery," whilst Volume V. will be greatly enlarged, and treat upon "The Theory of Machines."

Separate Contents and Index have been carefully arranged for each Volume. These enable students to find the details and pages where the different subjects are treated. The Author's system of Engineering Symbols, Abbreviations, and Index Letters have been printed at the beginning of each volume.

It is thus hoped, that the size and cost of each volume will suit the requirements of every Student of Engineering.

The Author has again to thank his chief assistant, Mr. John Ramsay, A.M.Inst.C.E., for his help, and his Publishers for the special care with which they have prepared these five volumes. The Author is specially indebted to Professor Alexander MacLay, B.Sc., C.E., and the Technical Publishing Company, Ltd., Manchester, for permission to abstract certain parts and figures from his excellent book on "*Loci* in Mechanical Drawing, including Point Paths in Mechanisms." These additions appear in Vol. V. upon the "Theory of Machines," along with the new Lectures dealing with the "Kinematics of Machinery."

The Author will feel much obliged to Engineers, Teachers, his B.Sc. and C.E. "Correspondence Students," also to any other Students of Engineering, for hints which may tend to further enhance the attractiveness and usefulness of these works.

ANDREW JAMIESON.

Consulting Engineer and Electrician,
16 Rosslyn Terrace, Kelvinside,
Glasgow, *September*, 1909.

INSTRUCTIONS FOR ANSWERING HOME EXERCISES.

(*As used in my Engineering and Electrical Science Correspondence System.*)

---

1. Use ordinary foolscap paper, and write on the left side *only*, leaving the facing page blank for my corrections and remarks.

2. Put the date of the Exercises at the left-hand top corner; your Name and Address in full, the name of the Subject or Section, as well as number of Lecture or Exercise, in the centre of the first page. The number of each page should be put in the right-hand top corner.

3. Leave a margin 1¼ inches wide on the left-hand side of each page, and in this margin place the *number* of the question *and nothing more.* Also, leave a clear space of *at least* 2 inches deep between your answers.

4. Be sure you understand *exactly* what the question requires you to answer, then give *all* it requires, but *no more.* If unable to fully answer any question, write down your own best attempt and state your difficulties.

5. Make your answers concise, clear, and exact, and always accompany them, if possible, by an *illustrative sketch.* Try to give (1) Side View, (2) Plan, (3) End View. Where asked, or advisable, give Sections, or Half Outside Views and Half-Sections for (1), (2), and (3).

6. Make all sketches large, open, and in the centre of the page. Do not crowd any writing about them. Simply print sizes and index letters (or names of parts), with a bold Sub-heading of what each figure or set of figures represent.

7. The character of the sketches will be carefully considered in awarding marks to the several answers. Neat sketches and "*index letters,*" having the first letter of the name of the part, will always receive more marks than a bare written description.

8. All students are strongly recommended to *first* thoroughly study the Lecture, *second* to work out each answer in scroll, and *third* to compare it with the question and the text-book in order to see that each item has been answered. The final copy should be done without *any aid or reference* to the scroll copy or to *any text-book.* This acts as the best preparation for sitting at Examinations.

9. Reasonable and easily intelligible contractions (*e.g.*, mathematical, mechanical or electrical, and chemical symbols) are permitted.

10. Each corrected answer which has the symbols R.W. marked thereupon must be carefully re-worked as *one* of the set of answers for the following week.

ANDREW JAMIESON.

15 ROSSLYN TERRACE, KELVINSIDE, GLASGOW.

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VOL. III.—Theory of Structures and Graphic Statics, with Applications to Roofs, Cranes, Beams, Girders, and Bridges.

VOL. IV.—Hydraulics.—Hydraulic and Refrigerating Machinery.

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MECHANICAL ENGINEERING SYMBOLS, ABBREVIATIONS, AND INDEX LETTERS

USED IN VOLUMES I. TO V.

Of Professor Jamieson's "Applied Mechanics."

Prefatory Note.—It is very tantalising, as well as a great inconvenience to Students and Engineers, to find so many different symbol letters and terms used for denoting one and the same thing by various writers on mechanics. It is a pity, that British Civil and Mechanical Engineers have not as yet *standardised* their symbols and nomenclature as Chemists and Electrical Engineers have done. The Committee on Notation of the Chamber of Delegates to the International Electrical Congress, which met at Chicago in 1893, recommended a set of "Symbols for Physical Quantities and Abbreviations for Units," which have ever since been (almost) universally adopted throughout the world by Electricians.* This at once enables the results of certain new or corroborative investigations and formulæ, which may have been made and printed anywhere, to be clearly understood anywhere else, without having to specially interpret the precise meaning of each symbol letter.

In the following list of symbols, abbreviations and index letters, the *first* letter of the chief noun or most important word has been used to indicate the same. Where it appeared necessary, the *first* letter or letters of the adjectival substantive or qualifying words have been added, either as a following or as a subscript or suffix letter or letters. For certain specific quantities, ratios, coefficients and angles, small Greek letters have been used, and I have added to this list the complete Greek alphabet, since it may be refreshing to the memory of some to again see and read the names of these letters, which were no doubt quite familiar to them when at school.

---

* These "Symbols for Physical Quantities and Abbreviations for Units" will be found printed *in full* in the form of a table at the commencement of Munro and Jamieson's *Pocket-Book of Electrical Rules and Tables.* If a similar recommendation were authorised by a committee composed of delegates from the chief Engineering Institutions, it would be gladly adopted by "The Profession" in the same way that the present work of "The Engineering Standards Committee" is being accepted.

TABLE OF MECHANICAL ENGINEERING QUANTITIES, SYMBOLS, UNITS AND THEIR ABBREVIATIONS.

(*As used in Vols. I. to V. of Prof. Jamieson's "Applied Mechanics."*)

Quantities.	Symbols.	Defining Equations.	Practical Units.	Abbreviations of the Practical Units.
FUNDAMENTAL.				
Length,	L, l	...	Yard, Foot, Inch,	yd. ft. in.
Mass,	M, m	...	Pound,	lb.
Time,	T, t	...	Second, Minute, Hour,	s. m. h.
GEOMETRIC.				
Surface,	S, s	$S = L^2$	Square foot, Square inch,	sq. ft. sq. in.
Volume,	V	$V = L^3$	Cubic foot, Cubic inch,	cb. ft. cb. in.
Angle, ∠	α, β θ, ϕ	$\alpha = \frac{\text{arc}}{\text{radius}}$	Degree, Minute, Second, Radian $= \frac{180°}{\pi}$	1° 1′ 1″ rn.
MECHANICAL.				
Velocity,	v	$v = \frac{L}{T}$	Foot per second,	$\frac{\text{ft.}}{\text{s.}}$
Angular velocity,	ω	$\omega = \frac{v}{L} = \frac{\theta}{t}$	Revs. per second, Revs. per minute, Radians per second,	r.p.s. r.p.m. ω
Acceleration,	a, g	$a = \frac{v}{T}$	Foot per sec. per sec.	$\frac{\text{ft.}}{\text{s}^2}$
Force,	F, f	...	Pound weight (gravitational unit),	lb. wt. (or lb.)
	W, w	$F = M a$	Poundal (absolute unit),	pdl.
Pressure (per unit area),	p	$p = \frac{F}{s}$	Pound per sq. inch,	lb. □″
Work,	$(W h)$	$W h = F L$	Foot-pound,	ft.-lb.
Potential energy,	E_P	$E_P = W h$	Foot-pound,	ft.-lb.
Kinetic energy,	E_K	$E_K = \frac{W v^2}{2 g}$	Foot-pound,	ft.-lb.
Power or activity,	HP	$\text{H.P.} = \frac{W h}{T}$	Horse power, Ft.-lb. per min., Ft.-lb. per sec.,	H.P. ft.-lb./m. ft.-lb./s.
Moment of inertia,	I	$I = M k^2$	... ...	lb.-ft.^{2}
Density,	ρ	$\rho = \frac{M}{V}$	Pound per cb. ft., Pound per cb. in.,	$\frac{\text{lb.}}{\text{ft.}^3}$ $\frac{\text{lb.}}{\text{in.}^3}$

OTHER SYMBOLS AND ABBREVIATIONS IN VOLS. I. TO V.

A for Areas.
B, *b* ,, Breadths.
C, *c*, *k* ,, Constants, ratios.
c.g. ,, Centre of gravity.
D, *d* ,, Diameters depths, deflections.
D_1, D_2, D_3 ,, Drivers in gearing.
E ,, Modulus of elasticity.
e ,, Velocity ratio in wheel gearing.
F_1, F_2, F_3 ,, Followers in gearing.
f_s, f_t ,, Forces of shear and tension.
H, *h* ,, Heights, heads.
H.P., h.p. ,, Horse-power.
B.H.P. ,, Brake horse-power.
E.H.P. ,, Effective ,,
I.H.P. ,, Indicated ,,
k ,, { Radius of gyration, or, Coef. of discharge in hydraulics.
N, *n* ,, Numbers—*e.g.*, number of revs. per min , number of teeth, &c.
P, Q ,, Push or pull forces.
R_1 R_2 ,, Reactions, resultants, radii, resistances.
s ,, { Seconds, space, surface. Displacement, distance.
S F ,, Shearing force.
T M ,, Torsional moment.
T R ,, Torsional resistance.
B M ,, Bending moment.
M R ,, Moment of resistance.
R M ,, Resisting moment.
T_d, T_s ,, Tensions on driving and slack sides of belts or ropes, &c.
W_L, W_T, W_U ,, Lost, total, and useful work.

x, *y*, *z* for Unknown quantities.
Z ,, Modulus of section.
Z_t ,, ,, tension.
Z_c ,, ,, compression.

Δ, δ, *d* for Differential signs which are prefixed to another letter; then the two together represent a very small quantity.
ε, e ,, Represents base of Naperian Logs = 2·7182; for example, $\log_e 3 = 1{\cdot}1$.
η ,, Efficiency.
λ ,, Length ratio of ship to model.
μ ,, Coefficient of friction.
π ,, Circumference of a circle ÷ its diameter.
ρ ,, Radius of curvature, radian.

Σ for Symbol for sum total of a number of quantities.
$\int_o^x$,, Sign of integration or summation between limits 0 and *x*.
∼ ,, Sign for the difference between two quantities.
□ ,, Sign for square—*e.g.*, 10 □″ = 10 square inches.
— ,, Sign over two letters, $\overline{PQ}$, for a force acting from P to → Q, means that they represent a vector quantity, which has (1) magnitude, (2) direction, (3) sense.
≧ ,, Sign for equal to or greater than.
≦ ,, Sign for equal to or less than.

GREEK ALPHABET.

Α	α	Alpha.	Ι	ι	Iota.	Ρ	ρ	Rho.
Β	β	Beta.	Κ	κ	Kappa.	Σ	σ or ς	Sigma.
Γ	γ	Gamma.	Λ	λ	Lambda	Τ	τ	Tau.
Δ	δ	Delta.	Μ	μ	Mu	Υ	υ	Upsīlon.
Ε	ε	Epsilon.	Ν	ν	Nu	Φ	φ	Phi.
Ζ	ζ	Zeta.	Ξ	ξ	Xi.	Χ	χ	Chi.
Η	η	Eta.	Ο	ο	Omicron.	Ψ	ψ	Psi.
Θ	θ	Theta.	Π	π	Pi.	Ω	ω	Omĕga.

VOLUME IV.

ON

HYDRAULICS AND HYDRAULIC MACHINERY.

LECTURE I.

HYDROSTATICS—HYDRAULIC MACHINES.

CONTENTS.—Hydraulics—Fluids—Viscosity—Transmission of Pressure by a Fluid—Pressure of a Heavy Fluid—Head—Pressure on an Immersed Surface—Examples I., II., III., and IV.—Centre of Pressure—Centre of Pressure on a Rectangle—Triangle—Circle—Example V.—Energy of Still Water—Common Suction Pump—Belt-driven Suction Pump—Example VI.—Air Pump—Single-acting Force Pump—Single-acting Force Pump with Ball Valves—Force Pump with Air Vessel—Continuous Delivery Pumps without Air Vessels—Double-acting Force Pump—Double-acting Circulating Pump—Worthington Steam Pump—Pulsometer Pumps—Roots' Blower—Bramah's Hydraulic Press—Examples VII. and VIII.—Hydraulic Flanging Press—Hydraulic Jack—Examples IX., X., and XI.—Hydraulic Bear—Lead-covering Cable Press—Hydraulic Accumulator—Example XII.—Hydraulic Cranes—Hydraulic Wall Crane—Movable Jigger Crane—Double Power Hydraulic Crane—Hydraulic Capstan—Questions.

Hydraulics.—In its widest sense, the term "Hydraulics" is given to the study of the mechanical properties of fluids and their application to practical purposes. In a more restricted sense, it refers to the science of the pressure and flow of water and their applications in engineering. It is divided into two sections:—**Hydrostatics**, the science of fluids at rest; and **Hydrokinetics**, the science of fluids in motion.

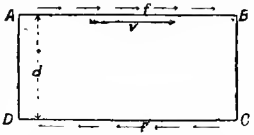
Fluids.—In many investigations it is necessary for simplicity to assume that we are dealing with a *perfect fluid;* that is, one which possesses the following property:—

DEFINITION.—**A fluid is a substance which offers no resistance to a continuous change of shape.**

There are two kinds of fluids—those which are practically incompressible, termed *liquids;* and those which are easily compressed, called *gases* and *vapours.* We know of no substance which completely fulfils the above definition; but water, many other liquids, and all gases, so nearly comply with it, that for many purposes we may, in practice, consider them as perfect fluids.

Viscosity.—Ordinary fluids, however, do offer some resistance to a change of shape, and the property in virtue of which they do so is called the *viscosity* of the fluid. A substance, such as syrup, which offers considerable resistance to a rapid change of form, but which goes on changing its shape so long as any deforming forces are applied to it, however small these forces may be, is usually called a *viscous fluid*. This term, strictly speaking, applies to all fluids, since all have some viscosity. It should, however, be noted that even the most mobile fluid will offer an appreciable resistance to a *sudden* deformation, because parts of it have to be set in motion and their inertia comes into play. This must not be confounded with their viscosity.

A solid body differs from a viscous fluid in that a small force produces in it a definite change of shape in a short time, and thereafter no further deformation takes place. Many solids, however, such as lead, tin, copper, and iron, when subjected to very great stresses, behave like viscous fluids, and keep *flowing* as long as the pressure is kept up. Even with very small forces, such apparently solid bodies as sealing wax and cobbler's wax, which fly to pieces when we subject them to a sudden force, such as a blow from a hammer, will gradually yield when sufficient time is allowed, and consequently they must be considered as very viscous fluids. For instance, a leaden bullet will sink in a thick piece of cobbler's wax and a cork will rise upwards through it, just as they would do through syrup or water, but they may take many months or years to do so.



Viscosity of a Fluid.

The viscosity of a fluid is measured by the shear stress required to deform it at the uniform rate of unit shear strain per unit time. Thus, if the figure represents a small portion of fluid and if a tangential stress f acts along A B and C D, the fluid will change its shape by the part A B moving along with a velocity v, relatively to the part D C.

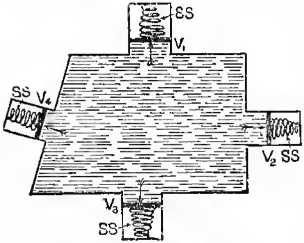
$$\text{Then,}\quad \left.\begin{array}{c}\textit{Shear strain produced}\\ \textit{in unit time}\end{array}\right\} = \frac{\textit{Velocity of}\ \mathrm{A}}{\textit{Distance}\ \mathrm{A\,D}}$$

$$\text{Or,}\quad \textit{Rate of shear} = \frac{v}{d} = \omega.$$

$$\therefore\quad \textbf{Coefficient of viscosity} = \mu = \frac{f}{\omega}. \quad . \quad . \quad . \quad . \quad (\mathrm{I})$$

When dealing with fluids at rest and in hydraulic machines, such as presses, cranes, &c., in which the motion of the liquid is comparatively slow, we need not take account of their viscosity; but when considering their flow through pipes and channels it becomes of great importance.

Transmission of Pressure by a Fluid.—Pascal's law, that "fluids transmit pressure equally in all directions," follows at once from our definition of a fluid. Thus, take a vessel filled with a fluid and fitted with several frictionless pistons which all have the same area and are held in place by springs. If we now apply an inward force through the spring to one of the pistons, say V_1, we shall find that each of the other pistons will be pushed outward with the same force. Had the pistons been of different areas we should have found the forces proportional to their areas, showing that the pressure per unit area is the same in all directions.

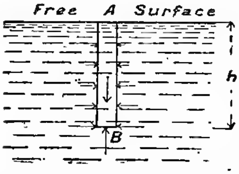


(HORIZONTAL SECTION.)

TRANSMISSION OF PRESSURE BY FLUIDS.

Another property following from our definition is that the pressure on any surface, real or imagined, is everywhere normal to that surface.

Pressure of a Heavy Fluid—Head.—Had the fluid in the above experiment been water or mercury and the pistons placed at different levels, we should have found that the pressure was not the same on all of them, but greatest on the lowest and least on the uppermost piston. This difference is due to the weight of the fluid. For example, if we have a quantity of liquid, the pressure at the bottom end B, of a vertical column A B, would be greater than at A; and, since the pressures round the sides of the column balance one another, the weight and the pressures on the ends must be in equilibrium. The difference of pressure is, therefore, equal to the weight of a cylinder of liquid whose length is A B and whose cross section has unit area. This will obviously be proportional to the length of the column A B; that is, to the difference of level.



PRESSURE OF A HEAVY FLUID.

In the figure, the upper surface is open to the atmosphere, and is, therefore, called the *Free Surface.* The pressure at A is atmospheric, but in connection with hydraulics it is customary to reckon this as the zero of reference, and when we speak of the pressure of a fluid we mean the excess of its pressure above that of the atmosphere.

Since we can have any pressure by taking A B of suitable length, we very often measure a pressure by the length of the vertical column of liquid which it will support—or, what is the same thing, which will produce the same pressure—and we shall term this length the *Head.* The column itself we shall refer to as the *Pressure Column.* With water each foot of head, and with mercury each inch, corresponds to nearly half a pound per square inch. The exact figures are:—

1 Foot of Water = 0·434 lb. per square inch.
1 Inch of Mercury = 0·491 ,, ,,

If h be the height of the free surface above the point we are considering, and w the weight of unit volume of the liquid, the pressure per unit area will be:—

$$p = w\,h. \qquad \ldots \ldots \ldots \qquad \text{(II)}$$

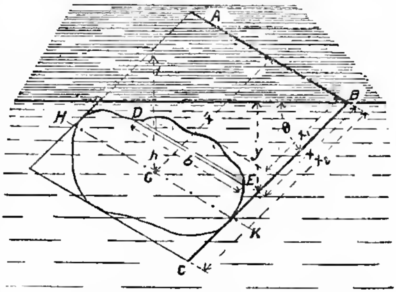
For fresh water, w may be taken as 62·42, or nearly $62\frac{1}{2}$ lbs. (1,000 ounces) per cubic foot, or 0·0361 lb. per cubic inch.

Pressure on an Immersed Surface.—If the above pressure be exerted on every unit of a surface whose area is a, the *total pressure* on that surface will be:—

$$\mathrm{P} = a \times p = a\,w\,h. \qquad \ldots \ldots \qquad \text{(III)}$$

This is also frequently called "the pressure," but it is usually quite clear whether the total pressure or the intensity of pressure is meant, although they are commonly denoted by the same term.

On a plane surface which is not level, the intensity of pressure is not the same for all parts. In such a case, we may find the total pressure as follows:—



Pressure on a Submerged Surface.

Let A B be the intersection of the plane of the submerged surface with the free surface of the water. Draw the line B C in that plane perpendicular to A B, and take D E, a very narrow strip or element of the surface, at right angles to B C and distant x from A B. A B and D E will be parallel since they are in one plane and both perpendicular to B C. Consequently, D E will be horizontal, and, therefore, the intensity of pressure over it will be uniform and equal to $w\,y$, y being the depth of the strip below the surface. Hence, if b be the breadth D E of the surface (*i.e.*, the length of the strip), and $d\,x$ that of the strip, the total pressure on the element will be $b\,d\,x \times w\,y$, or $b\,d\,x\,w\,x \sin\theta$ since $y = x\sin\theta$, if θ be the inclination of the plane to the horizontal. Now, we can split up the whole surface into a very large number of such elements and the total pressure on it will be the sum of all those on the elements :—

$$\therefore \qquad \mathrm{P} = w \sin\theta \int_{x_1}^{x_2} b\,x\,d\,x.$$

But $b\,x\,d\,x$ is the area of an element multiplied by its distance from A B, and, therefore, from the definition of the centre of gravity of a lamina, the integral is equal to the whole area a multiplied by the distance of its centre from A B. Let this distance be $\bar{x}$, and let $h = \bar{x}\sin\theta$, be the depth of the centre below the surface :—

$$\text{Then,} \qquad \mathrm{P} = w\sin\theta \times a\,\bar{x} = a\,w\,h. \qquad . \quad . \quad . \qquad (\mathrm{III}_a)$$

This is evidently the same pressure as if the surface were level and immersed at the same depth as its centre of gravity.

Example I.—A cylindrical tank, 6 feet in diameter and 10 feet deep, is filled with water; find the bursting pressure round the base of the tank, and the pressure on its base.

Answer.—The bursting pressure round the base is measured by the intensity of the fluid pressure on any small area of the curved surface infinitely near to the base. This pressure will be exactly equal to that on the base. Hence, the question resolves itself into finding the intensity of the pressure on the base.

$$\therefore \left.\begin{array}{l}\textbf{Bursting pressure}\\ \quad\textbf{round the base}\\ \quad\textbf{of tank}\end{array}\right\} = \textit{Pressure per square inch on base.}$$

$$\qquad ,, \qquad ,, \qquad = a\,h\,w.$$

$$\qquad ,, \qquad ,, \qquad = \frac{1}{144} \times 10 \times 62{\cdot}5 = 4{\cdot}34 \text{ lbs. per sq. in.}$$

$$\text{Again,} \quad \left.\begin{array}{r}\text{Total pres-}\\ \text{sure on base}\end{array}\right\} = \left\{\begin{array}{l}\textit{Area of base in sq. ins.} \times \textit{pressure}\\ \quad \textit{per sq. in.}\end{array}\right.$$

$$\text{Or,} \qquad ,, \quad ,, \qquad = \frac{22}{7} \times 36 \times 36 \times 4{\cdot}34 = 17{,}678 \text{ lbs.}$$

EXAMPLE II.—A circular water tank is 20 feet in diameter and 25 feet deep. It is constructed of 6 rings of cast-iron plates. Find the *total* stress on any vertical section of the bottom row of plates made by a plane passing through the axis of the cylinder, neglecting any assistance afforded by the flanges or connection with the bottom plate. (S. and A. Exam., 1888.)

ANSWER.—It has been proved in Lect. I., Vol. II., that when a cylindrical shell is subjected to internal fluid pressure, the *total* stress in the material along any section made by a plane containing the axis of the cylinder is equal to the total fluid pressure on either side of that part of the plane intercepted within the cylinder.

Hence, total stress in material of bottom row of plates = total fluid pressure on vertical plane through the axis of the cylinder at the bottom row of plates.

Since the breadth of each ring $= \frac{25}{6} = 4\frac{1}{6}$ ft.; therefore, depth of *c. g.* of bottom ring $= h = 25 - \frac{1}{2} \times 4\frac{1}{6} = 22{\cdot}9\dot{6}$ ft.

$$\therefore \left.\begin{array}{l}\text{Total stress in material}\\ \text{along section at bottom}\\ \text{row of plates}\end{array}\right\} = a\,h\,w,$$

$$\text{,,} \qquad \text{,,} \qquad = (20 \times 4\tfrac{1}{6}) \times 22{\cdot}9\dot{6} \times 62{\cdot}5 \text{ lbs.}$$

$$\text{,,} \qquad \text{,,} \qquad = 119{,}357 \text{ lbs.}$$

EXAMPLE III.—How is the pressure of water on a given area immersed in it ascertained? A water tank, 8 feet long and 8 feet wide, with an inclined base, is 12 feet deep at the front and 6 feet deep at the back, and is filled with water. Find the pressure in lbs. on each of the four sides, and on the base; water weighing $62\frac{1}{2}$ lbs. per cubic foot.

ANSWER.—The total fluid pressure on any area immersed in the fluid is given by the formula—$P = a\,h\,w$.

Where a = Area of surface exposed to the fluid pressure,
h = Depth of centre of gravity of immersed area below free surface of fluid,
w = Weight of a cubic unit of fluid.

The shape and dimensions of the tank will be readily seen from the figure.

(*a*) *To find the total pressure on the front* A B C D.

Here, $a = AD \times DC = 8 \times 12 = 96$ sq. ft.
$h = \frac{1}{2}$ depth $DC = 6$ ft. $w = 62\frac{1}{2}$ lbs. per cubic ft.

$$\therefore \text{ Pressure on front } ABCD = a\,h\,w,$$

$$\text{,,} \qquad \text{,,} \qquad = 96 \times 6 \times 62\tfrac{1}{2} = 36{,}000 \text{ lbs.}$$

(*b*) *To find the total pressure on the back* E F M N.

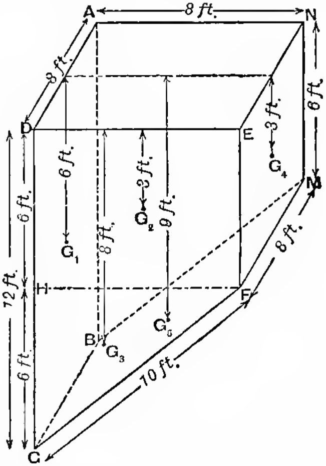
Here, $a = \text{F M} \times \text{M N} = 8 \times 6 = 48$ sq. ft.; $h = \frac{1}{2}\,\text{E F} = 3$ ft.

$\therefore$ **Pressure on back E F M N** $= a\,h\,w$,

„ „ $= 48 \times 3 \times 62\frac{1}{2} =$ **9,000 lbs.**

(*c*) *To find the total pressure on base* C B M F.

Before we can find the area of the base, we must know its length C F. From F draw F H parallel to E D, and therefore perpendicular to D C. Then C H F is a right-angled triangle whose sides are F H = E D = 8 ft., and H C = D C − D H = D C − E F = 6 ft.



Pressure on Sides of Tank.

$$\therefore \qquad \text{C F} = \sqrt{\text{H F}^2 + \text{H C}^2} = \sqrt{8^2 + 6^2} = 10 \text{ ft.}$$

$$\therefore \qquad a = \text{C F} \times \text{C B} = 10 \times 8 = 80 \text{ sq. ft.}$$

Again, the depth of the *c. g.* of the base C B M F is clearly—

$$h = \tfrac{1}{2}\,(\text{D C} + \text{E F}) = 9 \text{ ft.}$$

$\therefore$ Pressure on base C B M F $= a\,h\,w$,

„ „ $= 80 \times 9 \times 62\frac{1}{2} =$ **45,000 lbs.**

Missing Page

To find the depth of the centre of pressure we may proceed as follows:—

Referring to our former figure we see that the moment about A B of the pressure on the elementary strip D E is $w\,y \times b\,d\,x \times x$, or $w \sin\theta\, b\, x^2\, d\,x$. Hence the total moment is:—

$$M = w \sin\theta \int_{x_1}^{x_2} b\,x^2\,d\,x.$$

Now, $b\,x^2\,d\,x$ is the product of the area of an element into the square of its distance from A B, and consequently $\int_{x_1}^{x_2} b\,x^2\,d\,x$ is the second moment, or moment of inertia, of the area about A B. As shown in equation (III) of Lect. XII., Vol. I., it is, therefore, equal to $I + a\,\bar{x}^2$, where I is the moment of inertia about an axis H K, through the centre of gravity parallel to A B:—

$$\therefore \qquad M = w \sin\theta\,(I + a\,\bar{x}^2).$$

Again, the moment of the resultant must be the sum of the moments of its components. Let X be the distance of the centre of pressure from A B:—

Then, $$P\,X = M = w \sin\theta\,(I + a\,\bar{x}^2).$$

$$\therefore \qquad X = \frac{w \sin\theta\,(I + a\,\bar{x}^2)}{P} = \frac{w \sin\theta\,(I + a\,\bar{x}^2)}{w \sin\theta\, a\,\bar{x}}.$$

$$\therefore \qquad X = \frac{I + a\,\bar{x}^2}{a\,\bar{x}}. \qquad \text{(IV)}$$

That is, the distance of the centre of pressure from A B is the ratio of the second moment of the surface about A B to its first moment about the same axis.

If for I we write $a\,k^2$, k being the radius of gyration about the axis H K, and h for $\bar{x} \sin\theta$, we get:—

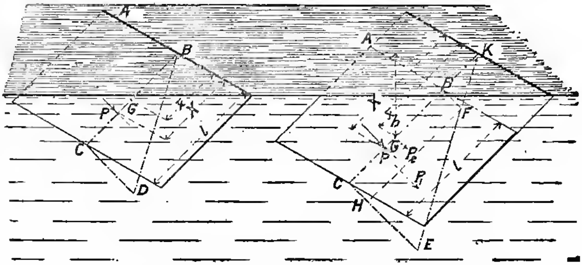
$$X = \frac{a\,k^2 + a\,\bar{x}^2}{a\,\bar{x}} = \frac{k^2 + \bar{x}^2}{\bar{x}} = \frac{k^2 \sin^2\theta + h^2}{h \sin\theta}. \qquad \text{(V)}$$

We shall now show how to apply these results to a few simple cases, and will also explain an easier method for special cases.

Centre of Pressure on a Rectangle.—First, consider a rectangle immersed with one edge in the surface. We find from Table II. in Lect. XII., Vol. I., that the value of k^2 for a rectangle is $\frac{1}{12} l^2$, where l is the length of the rectangle at right angles to the axis. Also, $\bar{x}$ will be $\frac{1}{2} l$:—

$$\therefore \qquad X = \frac{\frac{1}{12} l^2 + \frac{1}{4} l^2}{\frac{1}{2} l} = (\tfrac{1}{6} + \tfrac{1}{2}) l = \tfrac{2}{3} l. \quad . \quad . \quad . \qquad \text{(VI)}$$

If the rectangle be immersed further, until its centre is at a depth h, the top edge being kept horizontal, we do not get such a simple result, but it can be at once obtained for any given case from equation (V).



Centre of Pressure on a Rectangle.

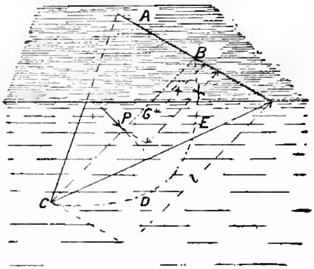
We can also obtain our result in the following manner:—At every point in B C—the central line of the rectangle—draw a line at right angles to it of such a length as to represent the whole pressure on a horizontal strip at that level. When the ends of these are joined we will have a triangle B C D, whose area represents the total pressure on the rectangle. The resultant pressure will pass through the centre of gravity of this triangle, and will, therefore, be two-thirds down from the vertex. Hence, the centre of pressure is distant two-thirds of the length of the rectangle from the top.

When the upper edge of the rectangle is below the surface, we obtain, instead of a triangle to represent the pressure, a trapezium B F E C, whose inclined sides, when produced, meet in the surface of the liquid. We ascertain the centre of pressure by finding the resultant of two forces, P_1 and P_2, the former of which is proportional to the area of the triangle F E H, and is two-thirds down from F, while the latter passes through the centre of the rectangle B F H C, and is proportional to its area. This may be done graphically as explained in Lecture V., Vol. III.

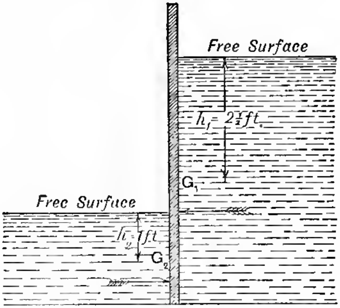
Centre of Pressure on a Triangle.—For a triangle with its base in the surface, $k^2 = \frac{1}{18} l^2$, and $\bar{x} = \frac{1}{3} l$:—

$$\therefore \qquad X = \frac{\frac{1}{18} \quad + \frac{1}{9} l^2}{\frac{1}{3} l} = (\tfrac{1}{6} + \tfrac{1}{3}) l = \tfrac{1}{2} l \quad . \quad . \qquad \text{(VII)}$$

This may also be proved geometrically. The intensity of pressure on a horizontal strip is proportional to its depth below the surface, while the length of the strip, and, therefore, its area, is proportional to its distance from the vertex C. Consequently, the whole pressure on each horizontal element will be proportional to the product $x\ (l - x)$ for elements of the same width. The area representing the pressure will, therefore, be a parabola, B E D C, passing through B and C, with its axis perpendicular to B C, and, consequently, the centre of pressure must be half way down.



Centre of Pressure on a Triangle.



Pressure on Sluice Gate.

If the vertex is in the surface, and the base horizontal, then $\bar{x} = \frac{2}{3}\, l$:—

$$\therefore \qquad X = \frac{\frac{1}{18}\, l^2 + \frac{4}{9}\, l^2}{\frac{2}{3}\, l} = \left(\tfrac{1}{12} + \tfrac{2}{3}\right) l = \tfrac{3}{4}\, l. \quad . \quad . \quad \text{(VIII)}$$

Centre of Pressure on a Circle.—The only other case we shall consider is that of a circle immersed vertically, with its centre at a depth h. Here $k^2 = \frac{1}{4} r^2$, and $\bar{x} = h$:—

$$\therefore \qquad X = \frac{\frac{1}{4} r^2 + h^2}{h}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(IX)}$$

When the circumference just touches the surface, $h = r$, and this becomes :—

$$X = \tfrac{5}{4}\, r = \tfrac{5}{8}\, d. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(X)}$$

Where r is the radius, and d the diameter of the circle.

Example V.—A sluice gate is 4 feet broad and 6 feet deep, and the water rises to a height of 5 feet on one side, and 2 feet on the other side. Find the pressure on the gate, and the centres of pressure.

Answer.—The net pressure on the sluice gate is evidently equal to the difference of the pressures on the two sides.

Total Pressure on Back $= a_1 h_1 w = (4 \times 5) \times 2{\cdot}5 \times 62{\cdot}5 = 3{,}125$ lbs.

,, ,, *Front* $= a_2 h_2 w = (4 \times 2) \times 1 \times 62{\cdot}5 = 500$,,

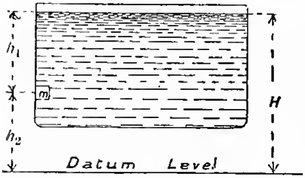
∴ **Net pressure on gate = 2,625 ,,**

The centre of pressure on the upper side is one-third of 5 feet, or 1 foot 8 inches, up from the bottom, and on the lower side, a third of 2 feet, or 8 inches. To find the resultant centre of pressure take moments about the bottom of the gate. Then, if this centre be distant x inches from the bottom :—

$$2{,}625 \times x = 3{,}125 \times 20 - 500 \times 8 = 62{,}500 - 4{,}000.$$

$$\therefore \quad x = \frac{58{,}500}{2{,}625} = 22{\cdot}3 \text{ ins.} = 1 \text{ ft. } 10{\cdot}3 \text{ ins.}$$

Energy of Still Water.—When water is at rest it possesses potential energy in virtue of its position and of its pressure. Consider a tank filled with water, and imagine a small mass m of the water to escape from the tank. This mass will not only lose potential energy through falling to a lower level, but it could also do work because of the pressure of the rest of the water pushing it away. It is convenient to assume some datum level at which we take the energy of position as zero.



Energy of Still Water.

Let H = Height of free surface above the datum level.
,, h_1 = Height of free surface above m.
,, h_2 = Height of m above datum level.
,, g = Acceleration due to gravity.
,, ρ = Density of fluid = mass of unit volume.
And, w = Weight of unit volume of fluid = ρg.

Then the work done in forcing out the mass m is :—

$$\textit{Energy of Pressure} = \textit{Volume} \times \textit{Pressure}.$$

$$,, \quad ,, \quad = \frac{m}{\rho} \times w h_1 = m g h_1.$$

And, *Energy of Position* $= m g h_2$.

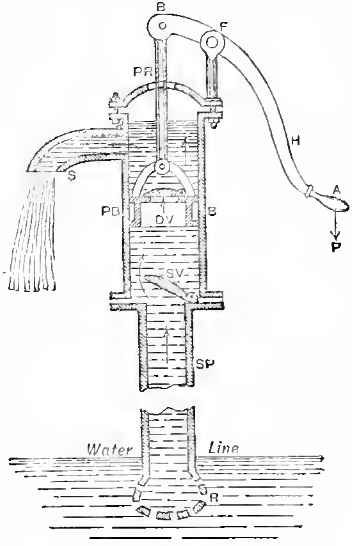
∴ *Total Energy* $= m g (h_1 + h_2) = m g \mathrm{H}$.

Or, for a unit mass :—

$$\textbf{Energy per unit mass} = g (h_1 + h_2) = g \mathrm{H} \quad . \quad \text{(XI)}$$

This is constant for all parts of a homogeneous fluid at rest, and H may be called the *total head* of the water in the tank.

Common Suction Pump.—This consists of a bored cast-iron barrel P B, terminating in a suction pipe, S P, fitted with a perforated end or rose R, which dips into the well from which the water is to be drawn. The object of the rose is to prevent leaves or other matter getting into the pump, that might clog and spoil the action of the valves. At the junction between the barrel and suction pipe there is fitted a suction valve S V, of the hinged clack type faced with leather. The piston or bucket B is worked up and down in the barrel of the pump by a force P, applied to the end of the handle H. This force is communicated to it through the connecting link of the hinged piston-rod, P R. In the centre and at the top of the bucket is fixed the clack delivery valve D V, which is also faced with leather in order to make it water-tight. The bucket is sometimes packed with leather; but, in the present instance, a coil of tightly woven flax rope is wrapped round the packing groove.



Common Suction Pump.

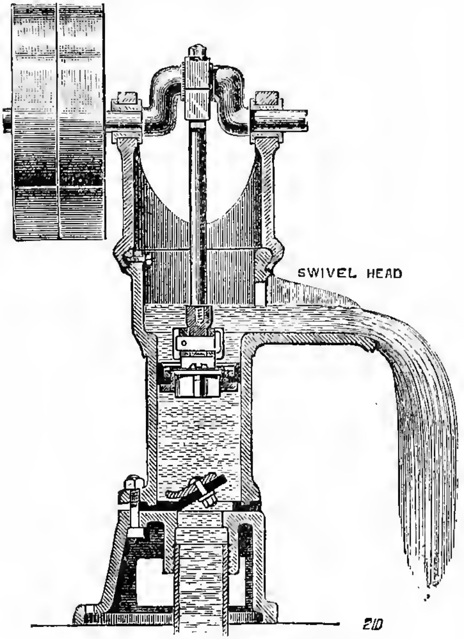
Action of the Suction Pump.—(1) Let the barrel and the suction pipe be filled with air down to the water-line, and let the bucket be at the end of the down stroke. Now raise the bucket to the end of the up stroke by depressing the pump handle. This tends to create a vacuum below the delivery valve; therefore, the air which filled the suction pipe opens the suction valve, expands, and fills the whole volume of the barrel. Consequently, according to Boyle's law, its pressure must be diminished in the *inverse ratio* to the enlargement of its volume. This enables the pressure of the atmosphere to force a certain quantity of water up the suction pipe, until the weight of this column of water and the pressure of the air between the suction and delivery valve, balance the pressure of the outside atmosphere.

(2) In pressing the bucket to the bottom of the barrel by ele-

vating the handle, the suction valve closes and the delivery valve opens, thereby permitting the compressed air in the barrel to escape through the delivery valve into the atmosphere.

(3) Raise and depress the piston several times so as to produce the above actions over again, and thus gradually diminish the volume of the air in the pump to a minimum. Then water will have been forced by the pressure of the atmosphere up the suction pipe and into the pump, if the bucket and the valves are tight, and if the delivery valve when at the top of its stroke be not more than the height of the hydro-barometric column above the water line of the well.*

(4) The bucket now works in water instead of in air; in fact, the machine passes from being an air-pump to being a water one. During the down stroke of the piston water is forced through the delivery valve, and during its up stroke, this water is ejected through the spout; at the same time, more water is forced up through the suction pipe and valve to fill the vacuum created by the receding piston. In the case of a common suction pump water is therefore discharged *only* during the up stroke of its piston.



Belt-driven Suction Pump.

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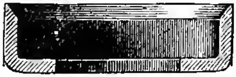
* Theoretically, such a pump should be able to lift water from a depth of 34 feet below the highest part of the stroke of the delivery valve, but practically, owing to the imperfectly air-tight fitting of the piston and the valves, it is not used for withdrawing water from wells more than 20 to 25 feet below this position of the delivery valve. In fact such a pump frequently requires a bucket or two of water to be poured into it above the delivery valve in order to make it work at all, if it should have been left standing for some time without being worked.

Belt-driven Suction Pump.—When we have to raise a considerable quantity of water during a long time, then it becomes advisable to apply power derived from some prime mover. The foregoing figure illustrates an ordinary suction pump driven by a belt, with fast and loose pulleys, crank shaft, and connecting-rod. Here the suction valve and the piston are faced with leather or india-rubber, whilst the bucket valve is made of brass and ground to fit its seat. The upper portion of the pump is fitted with a swivel head, so that the pulleys may be placed fair in line with the driving pulleys.



LEATHER PACKING FOR PISTON OF SUCTION PUMP.

EXAMPLE VI.—Given two simple bucket pumps, each having a stroke of 1 foot, and cross area of bucket, 20 square inches. Suppose everything perfectly air tight, and the supply pipe 20 square inches area in one case, and 10 square inches area in the other. Neglecting friction, you are to compare the tensions on the pump rods at ends of first up stroke in each case, supposing the bucket to be 24 feet above free surface of the water in the well when at the bottom of its stroke. The supply pipe reaches to the under surface of bucket when the latter is at the bottom of its stroke.

ANSWER.—

Let p = Pressure in lbs. per *sq. ft.* on *under* surface of bucket at end of first up stroke.

„ a = Area of bucket $= \frac{20}{144}$ sq. ft.

„ h = Height of water barometer = 34 ft.

„ w = Weight of 1 cub. ft. of water = $62\frac{1}{2}$ lbs.

„ x = Height water rises in suction pipe for first up stroke.

FIRST CASE.—*Lifting or suction pipe having an area equal to that of the bucket.*

Then at end of first up stroke, we get:—

Pressure of air on upper surface of bucket, ... } = *Atmospheric Pressure*

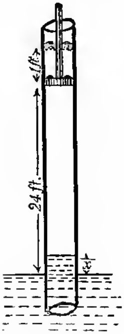
„ „ $= a\,h\,w$ *lbs.* . . . . . (1)

Pressure on under surface of bucket, $= p\,a = $ { *Atmos. pressure − pressure of small column,* x, *of water*

„ „ $= a\,(h - x)\,w.$

And, $p = (h - x)\,w.$ . . . . . (2)

Since there is exactly the same qnantity of air between the bucket and the surface of the water in the pipe at the end of the stroke as there was before the stroke commenced, we may apply Boyle's Law to determine the *volume* of air under the bucket at end of stroke.



FIRST CASE.

$$\therefore \qquad p\,a \times (25 - x) = a\,h\,w \times 24.$$

Substituting, $p = (h - x)\,w$, from equation (2), we get:—

$$(h - x)\,(25 - x) = 24\,h.$$

Since, $h = 34$ ft., we get, by substitution, and multiplication:—

$$x^2 - 59\,x + 34 = 0.$$

$$\therefore \qquad x = \frac{59 \pm 57{\cdot}83}{2}\text{ ft.}$$

The minus sign in the numerator of the fraction on the right-hand side of this equation is the only one admissible.

$$\therefore \qquad x = \frac{1{\cdot}17}{2} = {\cdot}58\text{ ft. } or = 7\text{ inches, nearly.}$$

Hence,

$$\text{Tension in pump rod} = \left\{\begin{array}{l} \textit{Press. on upper surface of bucket} \\ \quad - \textit{Press. on under surface.} \end{array}\right.$$

$$\text{,,} \qquad \text{,,} \qquad = a\,h\,w - a\,(h - x)w = a\,x\,w.$$

$$\text{,,} \qquad \text{,,} \qquad = \frac{20}{144} \times {\cdot}58 \times 62{\cdot}5 = 5{\cdot}04\text{ lbs.}$$

SECOND CASE.—*Lifting or suction pipe having an area equal to half that of the bucket.*

The symbols denoting the same quantities as before, we get:—

Pressure of air on upper surface of bucket $= a\,h\,w$ lbs. . . (3)

Pressure on under surface of bucket $= p\,a$

,, ,, $= a\,(h - x)\,w$. . . (4)

Vol. of air between bucket and surface of water at beginning of stroke . . $= \frac{1}{2}\,a \times 24$

,, ,, $= 12\,a$ *cub. ft.*

Vol. of air between bucket and surface of water at end of up stroke . . . $= \frac{1}{2}\,a \times (24 - x) + a \times 1$

,, ,, $= \frac{1}{2}\,(26 - x)\,a$ *cub. ft.*

∴ By Boyle's Law, we get :—

$$(h - x) w \times \tfrac{1}{2} (26 - x) a = h w \times 12 a.$$

Substituting $h = 34$, and simplifying, we get:—

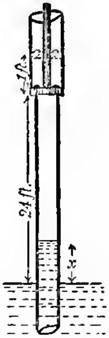
$$x^2 - 60 x + 68 = 0$$

$$\therefore \qquad x = 1{\cdot}15 \text{ ft. } or = 13{\cdot}8 \text{ inches, nearly.}$$

Hence,

Tension in pump rod $= a x w.$

$$\text{,, } \quad \text{,, } \quad = \frac{20}{144} \times 1{\cdot}15 \times 62{\cdot}5 = 10 \text{ lbs.}$$



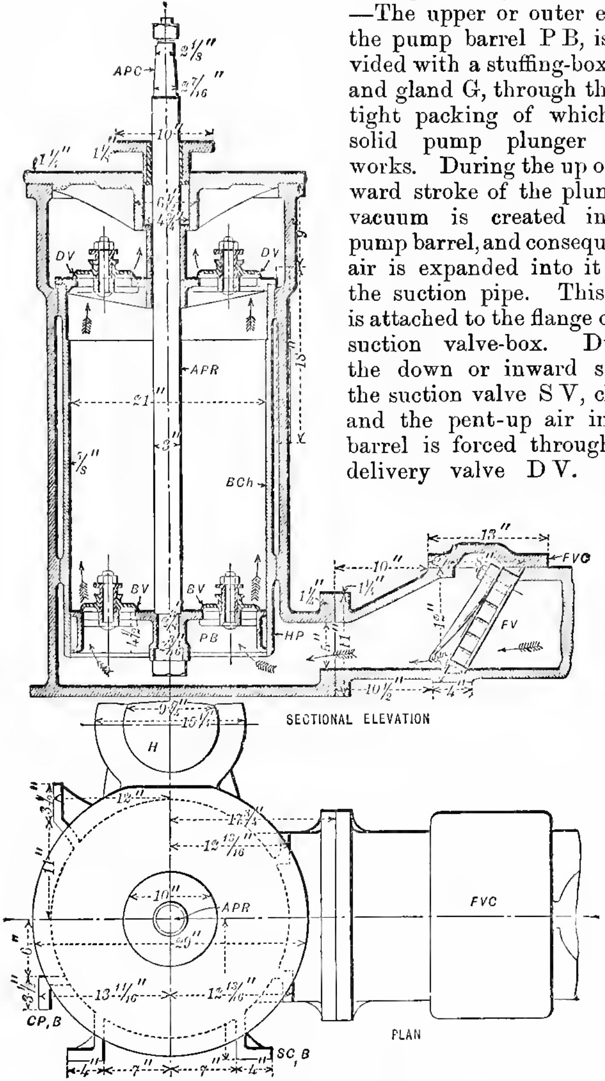
SECOND CASE.

Air Pump.—The figure on next page is a sectional elevation and outside plan of the air pump for the 1500 horse-power compound engines of the S.S. "St. Rognvald," which are fully described in the Author's *Text-Book on Steam and Steam Engines.* During the up stroke of the pump bucket P B, condensed steam and vapour are drawn from the surface condenser through the foot valve F V, into the space below the bucket, whilst any water and vapour that may have been lying above it, are forced through the delivery valves D V, into the hot well H. During the down stroke of the bucket, the water and vapour below it pass upwards through the bucket valves B V, into the space left by the descending bucket ; at the same time, the foot and delivery valves automatically close on their seats. These actions take place in succession during each up and down stroke of the air pump-rod A P R, which passes through an air-tight stuffing box, and is linked to the piston-rod crosshead of the high-pressure cylinder by short connecting-rods and side levers.

The object of placing the delivery valves on the top of the air-pump barrel in addition to the ordinary bucket valves, is to cause a vacuum to be produced above the latter during the down stroke of the bucket, and thus facilitate their opening, as well as to give the vapour from the condenser a free space between these two sets of valves into which it can expand.

The cast-iron barrel of the air pump is lined with a truly bored brass chamber B Ch, the pump bucket is rendered tight by hemp rope packing H P, the foot valve is readily inspected or removed by unbolting or lifting the foot valve cover F V C, whilst the whole is bolted securely to the surface condenser bracket S C_1 B, and to the circulating pump bracket C P_1 B.

Single-acting Force Pump.—The upper or outer end of the pump barrel P B, is provided with a stuffing-box S B, and gland G, through the air-tight packing of which the solid pump plunger P P, works. During the up or outward stroke of the plunger a vacuum is created in the pump barrel, and consequently air is expanded into it from the suction pipe. This pipe is attached to the flange of the suction valve-box. During the down or inward stroke the suction valve S V, closes, and the pent-up air in the barrel is forced through the delivery valve D V. This

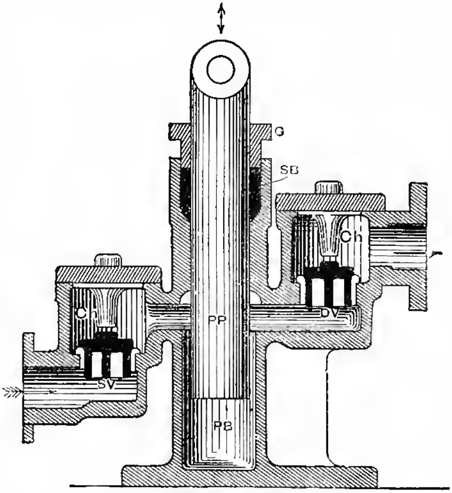


Air Pump for a Marine Engine.

action goes on precisely in the manner just explained in the case of the suction pump, until the water rises into the barrel. Then the inward stroke of the plunger drives through the delivery valve to any desired height or against any reasonable back pressure, as in the case of a feed pump for a steam boiler.

Both the suction and the delivery valves are made of brass, and fit accurately into their brass seats. The covers to the valve chests are provided with checks Ch, to prevent the valves from rising more than the distance required to pass the water freely through them.*

The eye of the plunger may be attached to a connecting-rod actuated by a hand lever, as in the case of the common suction pump, or it may be worked from an eccentric or crank revolved by a steam engine or other motor. By whichever way it is worked, the force applied to the plunger must be sufficient to overcome the friction between the plunger and the packing, the resistance due to sucking the water from the source of supply, and of driving the same up to the place where it is delivered.



SINGLE-ACTING FORCE PUMP.

As in the case of the suction pump, the water is only delivered during each alternate stroke of the plunger, and, consequently, in an intermittent or pulsating fashion.

Very often three pumps of this kind are combined in one, each plunger being driven by a separate crank on a common shaft, and the cranks making angles of 120° with each other. Such an arrangement is called a *three-throw pump*, and gives a very steady stream.

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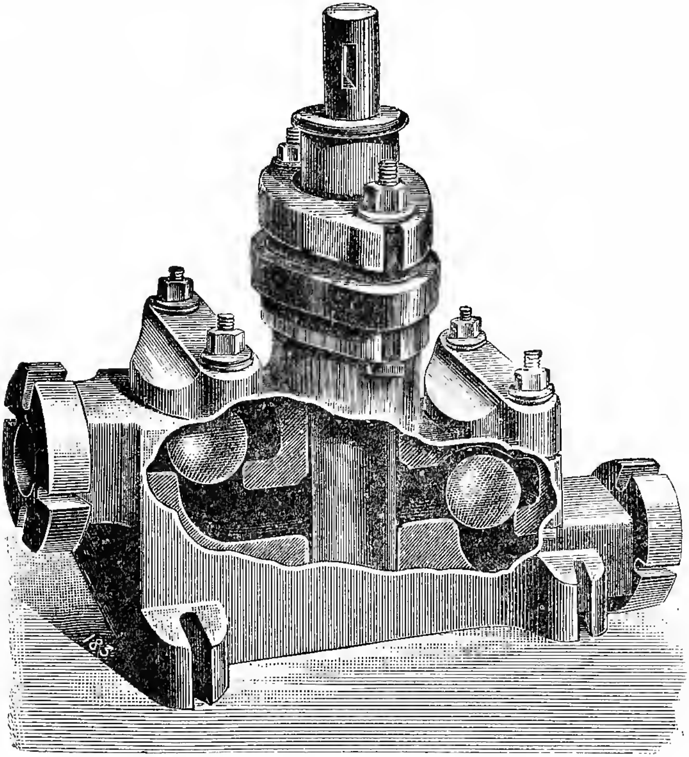
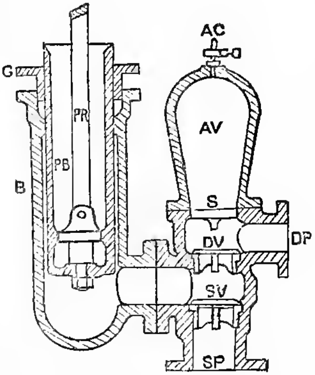
* If d be the diameter of the bore of the valve seat, and h the required lift of the valve to give an opening equal in area to that bore, then h must be quarter of d. For the area of bore $= \frac{\pi d^2}{4}$ and the equivalent area of the valve opening $= \pi d h$. $\therefore \frac{\pi d^2}{4} = \pi d h$. Or, $h = \frac{d}{4}$.

Single-acting Force Pump with Ball Valves.—The following illustration is a simple modification of the previous one, wherein ball valves are substituted for the common-circular three feathered type. The right-hand side forms the suction and the left-hand the delivery side. All the parts are made extra thick and strong to resist shocks and vibrations, and most of the bolt holes have been cored to the outside of the flanges for the purpose of facilitating rapid connection and disconnection. This form of pump is much used for forcing feed-water into steam boilers, &c.



Single-acting Force Pump with Ball Valves.

Force Pump with Air Vessel.—In the accompanying figure we have an illustration of a force pump with both the suction and the delivery valves placed on one side of the pump barrel and then surmounted by an air vessel. The plunger, instead of being solid, as in the previous cases, is made up of a hollow trunk or barrel, with a connecting-rod fixed to an eye-bolt at its lower end.

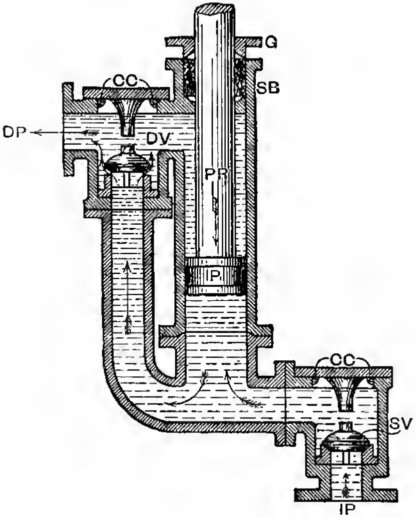


Force Pump with Air Vessel.

Action of the Air Vessel.—During the inward or delivery stroke of the plunger barrel P B, part of the water, which is forced from the barrel B, goes up through the delivery valve D V, into the delivery pipe D P, and the remainder enters the air vessel A V, and consequently compresses the air therein. During the outward or non-delivery stroke of the plunger the compressed air in the air vessel presses the rest of the water into the delivery pipe. In this simple way a continuous flow of water is maintained in the delivery pipe, and with far less shock, jar, and noise than in the previous cases. Where very smooth working is required, an air vessel is also put on to the suction pipe S P. Should the air in the air vessel become entirely absorbed by the water, the fact will be noticed at once, by the noise and the intermittent delivery. Then, the pump should be stopped, the air cock A C opened, and the water run out. When the air vessel is again full of air, the air cock should be shut and the pump restarted.

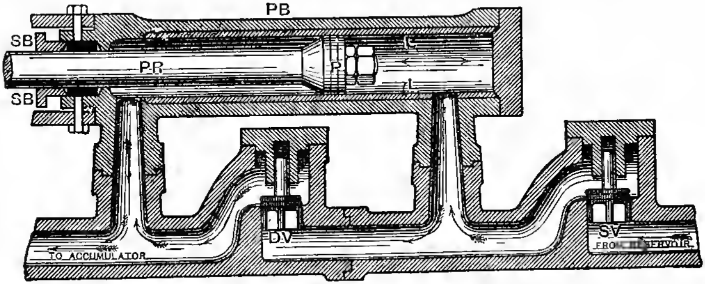
Continuous-delivery Pumps without Air Vessels.—A fairly continuous delivery of water may be obtained by making the plunger of the piston form, and the pump-rod exactly half its area; for here, during the down stroke, half the water expelled by the piston P, from the under side of the pump barrel goes up the delivery pipe D P, and the other half is lodged above the piston, to be in turn sent up the delivery pipe during the up stroke. Where very high pressures are required, such as in the filling of an accumulator ram, pumps working on this principle, but of the following form, are frequently used. The action is precisely the same as in the one just described, and the same index letters have been used, so that the student will have no difficulty in understanding the figure. The directions of motion of the piston and of the ingoing and outflowing water have been marked by straight and feathered arrows respectively.

With accumulators, and for other kinds of high-pressure work, it is not advisable to use air vessels, because you cannot prevent the water which enters them absorbing air and carrying the same with it to the hydraulic machines where its presence would be most objectionable. If 750 to 1000 or more lbs. pressure per square inch be generated, then you would require a very large and strong air vessel before it could be of any service. If a pressure of only 750 lbs. per square inch were used, then, since the normal pressure of the atmosphere is 15 lbs. per square inch, the air in the air vessel would be compressed, in accordance with Boyle's law, to $\frac{15}{750}$, or $\frac{1}{50}$ of its original volume. Consequently, with an air vessel of 50 cubic feet internal capacity, there would be only 1 cubic foot of air in it, when the pump was in full action.



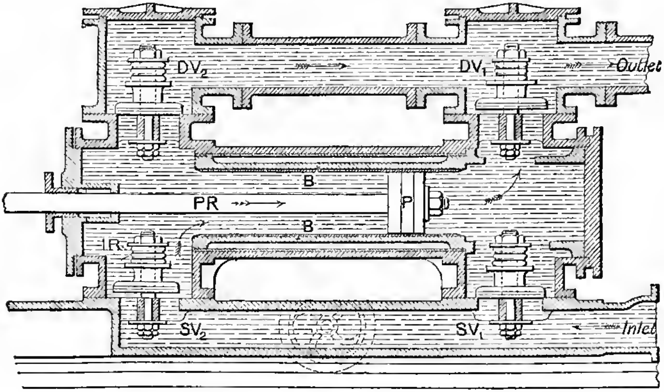
Continuous-delivery Force Pump without an Air Vessel.

Double-acting Force Pump.—The pumps which we have hitherto considered are all single-acting, in the sense that they do not both suck and discharge water during



Continuous-delivery Force Pump as used in Connection with the Armstrong Accumulator.

each stroke. This can, however, be accomplished by having two sets of suction and delivery valves placed at each end of the pump barrel, as shown by the accompanying figure. Here, during the outward stroke of the piston the pump draws water from the source of supply through the inlet pipe and suction valve $S V_1$, while, at the same time, the piston forces water in front of it through the delivery valve $D V_2$, and outlet pipe. During the inward stroke, suction takes place through $S V_2$ and discharge through $D V_1$, all as clearly shown by arrows in the drawing.

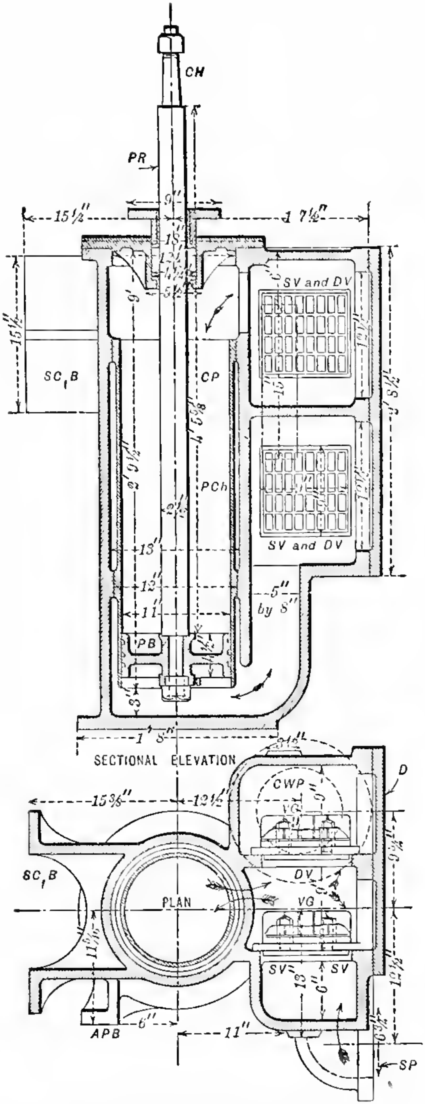


Double-acting Force Pump.

The valves are provided with india-rubber cushions I R, to ease the shock and minimise the jarring noise due to their reaction and natural reverberation when they are suddenly opened and closed.

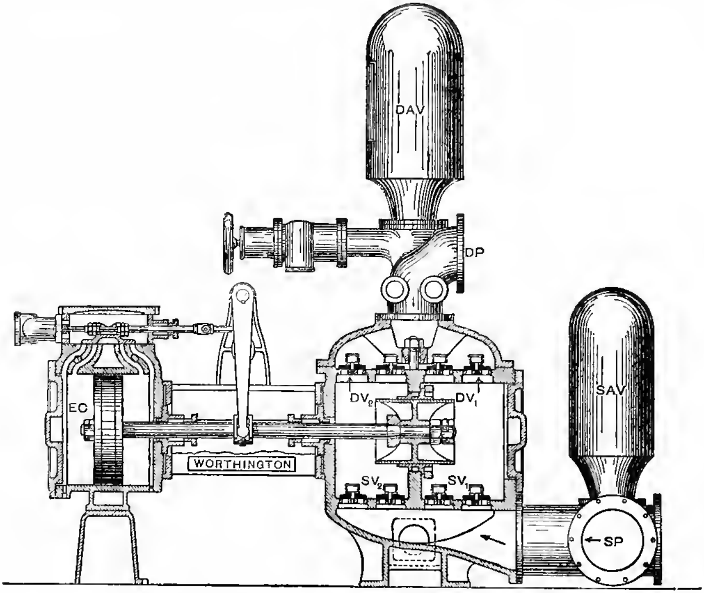
Double-acting Circulating Pump.—The following figure is a sectional elevation and plan of the circulating pump for the same marine engines as the previously described air pump. During the upstroke of the piston or pump bucket P B, water is drawn from the sea through the suction pipe S P, and the lower suction valves S V, into the lower part of the pump chamber P Ch. At the same time, the water from the top part of the chamber is forced up through the upper delivery valves D V, along the circulating water pipe C W P, into the surface condenser tubes, and from thence into the sea. During the down-stroke of the piston, the water which had previously entered by the bottom of the pump chamber is forced through the lower delivery valves D V, into the condenser tubes and sea, and at the same time, more water is taken into the top part of the pump chamber from the sea through the upper suction valves S V. These double actions take place in the manner described during each stroke, so that a stream of cold water is kept flowing through the condenser tubes, in order to maintain a vacuum during the exhaust of the steam from the low-pressure cylinder on to the outside of the cooled condenser tubes.

The cast-iron pump barrel of the circulating pump is lined with a truly bored brass pump chamber. The pump piston is also made of brass, and is rendered sufficiently water-tight by the simple device of turning three grooves in its outer cylindrical surface. The pump-rod passes through a water-tight gland and stuffing-box, and is connected to the reciprocating piston-rod cross-head by links and side levers, in the same way as the air pump-rod. The whole is securely bolted to the surface condenser and air pump brackets.



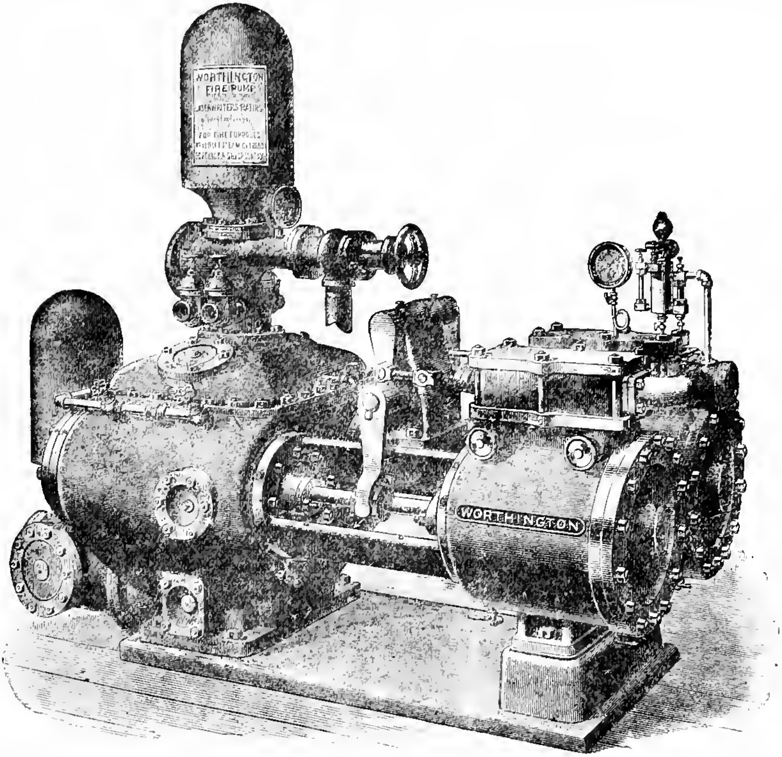
Double-acting Circulating Pump for a Marine Engine.

Worthington Steam Pump.—The perspective and sectional views of this well-known pump, together with the following description, will serve to explain the construction and action of one of the best examples of the duplex class of steam pumps for feeding boilers, working accumulators, and hoists. It is termed a duplex pump, from the fact that it consists of two steam and two water cylinders placed side by side. The pumps draw water from the suction pipe S P, and are in this case safeguarded from shock on the suction side by an air vessel S A V. The water is admitted through the suction valves $S\ V_1$, $S\ V_2$, at each stroke respectively, and delivered by the valves $D\ V_1$, $D\ V_2$, into the discharge pipe D P, under the smoothing action of the discharge air vessel **D A V**.



Vertical Section of the Worthington Steam Pump.

The steam pistons and the pump plungers are directly connected together by a piston-rod, and give a swinging motion to the intermediate long levers L, which are attached by two separate spindles to two shorter levers which work the slide valve spindles. Whenever one of the steam pistons moves towards either end of its stroke the other piston is approaching the opposite end of its stroke, and by the combination of levers, piston-rods, and spindles the slide valve of the one steam cylinder is actuated by its neighbour. The slide valves have neither lap nor lead, but immediately the piston of one cylinder covers one or other of the inner exhaust ports the steam in that cylinder is cushioned, and thus the pistons are prevented from striking their cylinder covers. Each piston as it reaches the end of its stroke automatically waits for its slide valve to be moved by the other piston-rod before it makes a return stroke. By this arrangement, the pump valves have time to close properly on their seats, and a natural smooth motion of the whole of the working parts takes place. There are no dead points in this form of duplex pump, consequently it is always ready to be started either



Perspective View of the Worthington Steam Pump.

by the opening of the stop valve or by the automatic action of a float connected to the throttle valve by a chain or rope.

Pulsometer Pumps.—The very first steam pump, which was invented by Thomas Savery in 1698, had no working parts except the valves. This type has been revived for certain kinds of work in pumps of the pulsometer class, of which Bailey's "Aqua Thruster" is a good example. It consists of two long chambers, in each of which there is a valve opening upwards at the bottom, and one opening outwards at the side. At the top junction between these two chambers there is a flap valve which can put either in communication with a steam pipe while the other is shut off therefrom.

Now, suppose the right-hand chamber to be full of water while the left one is full of steam, and that the upper valve is in the position shown. Steam will enter the right-hand chamber and force the water out through the delivery valve at the side. At the same time the steam in the left compartment will be condensing, and water will therefore rise into it through the bottom valve, provided the apparatus be not too far above the free surface of the water. The inertia of this water will cause it to continue in motion after all the steam is condensed, and it will therefore compress the air that remains to a sufficient extent to shift over the valve to the other side.* If there is no air, then the water itself will strike the valve and knock it over to the other side. The conditions of the chambers are now interchanged. Water will be forced out from the left one, and fresh water will rise into the other, and the process begins again.

A large loss occurs in this kind of pump through the condensation of steam during the down stroke of the water, and also owing to the fact that the steam is used non-expansively. To reduce the former loss little cocks open into the top of the chambers and admit a little air during the time there is a vacuum inside. This air prevents the steam from coming so quickly into contact with the water as it otherwise would do, and thus reduces the loss during admission. A slight escape of steam takes place

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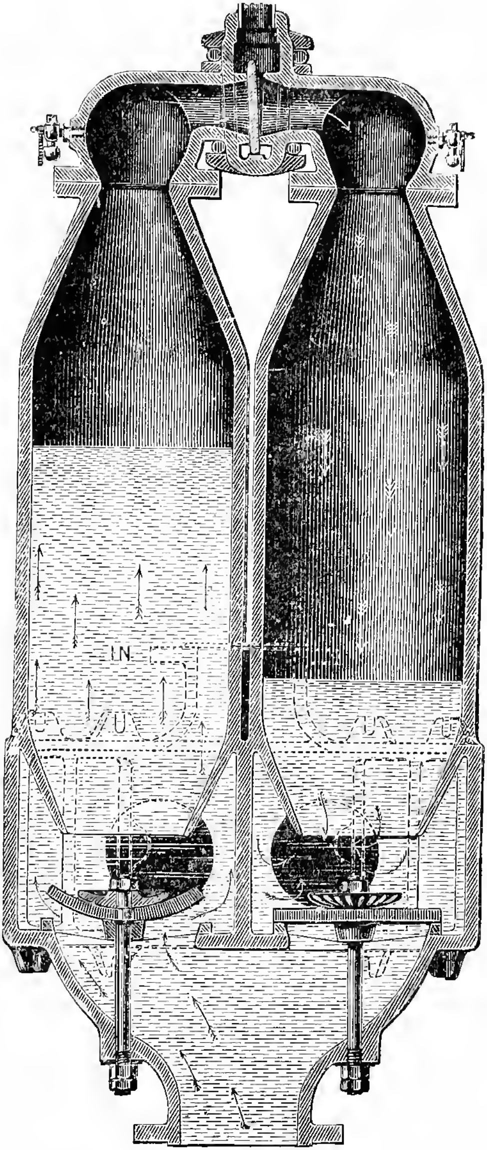
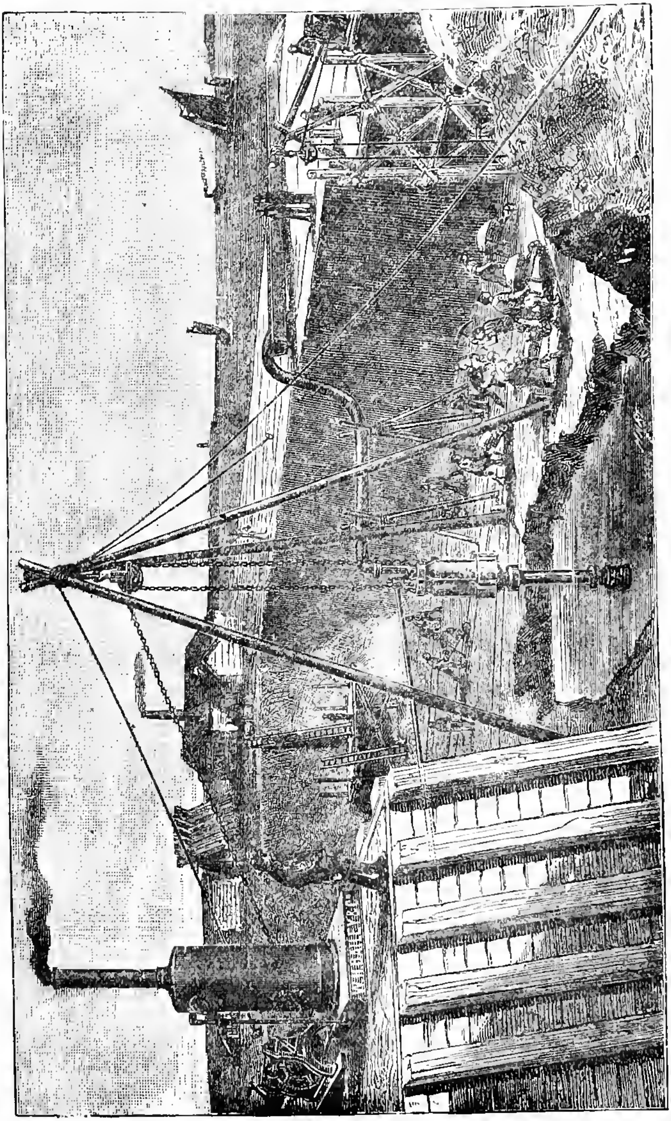
* This is not the common explanation of the working of the pulsometer valve. It is—"As soon as the water is lowered below the upper surface of the delivery valve, steam blows through with some violence and causes a commotion and a rapid condensation in the chamber. The valve is then *drawn* to the right-hand side." This is quite wrong. The valve can only be shifted by being *pushed*, owing to the pressure on the closed side becoming greater than that on the other side, and it is difficult to see how the pressure in a chamber in direct communication with the boiler can become less than that in one where the steam is already all condensed, and where the pressure is considerably below that of the atmosphere. Besides, it is probable that in steady working the water never gets as low as the delivery valves.



Pulsometer Pump by W. H. Bailey & Co., Ltd., Manchester.

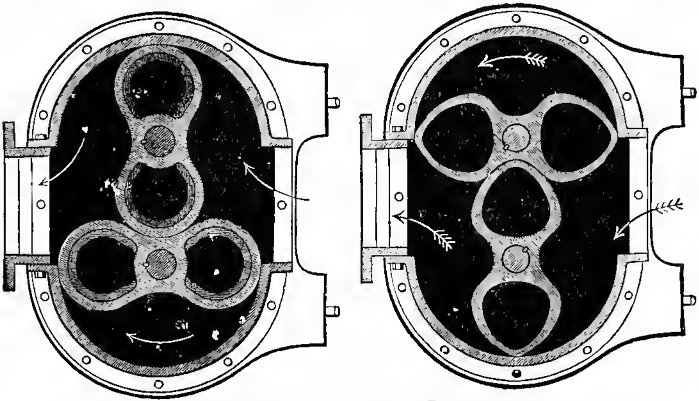


THE "AQUA THRUSTER" IN OPERATION.

through these cocks, as they are always kept open; but as their bore is so very small, the loss is less than the gain. To diminish the latter loss an extra self-acting valve, called the "grel," has been added to some forms of pulsometer with the intention of cutting off the steam earlier, and then using it expansively.

Pumps of this class are exceedingly handy for dealing with dirty water and for temporary purposes owing to their simplicity, few working parts, and the ease with which they can be erected. It is sufficient to suspend them by a chain and connect them by a pipe to a portable boiler. A suction pipe projects down below the water surface, and a flexible hose pipe will carry off the discharged water. The full page illustration shows the "Aqua Thruster" in use for pumping water from a dock during its construction.

Roots' Blower.—A form of rotary pressure pump, known as Roots' Blower, is used for obtaining a blast of air at a moderate pressure, and for pumping liquids. Two vanes rotate inside a closed casing, and sweep the fluid round with them. They are connected by spur wheels outside so as to be always at right angles to each other, and they have such a shape that practically nothing is carried backwards at the central part of the machine. They can produce a higher pressure than an ordinary blowing fan, and are handier for many purposes than a blower of the cylinder and piston type. The student should note that this is not a centrifugal pump or fan, although there are no reciprocating parts, but simply a rotary form of pressure pump.



Two Forms of Roots' Blower.

If a fluid be forced through this machine, then it will cause the vanes or teeth to rotate; hence it will work as a motor, and there-edges of the leather collar against the ram, and the other edge against the side of the bored cavity in the neck of the cylinder, with a force directly proportional to the pressure of the water in the cylinder. By this simple automatic action, the greater the pressure in the cylinder the tighter does the leather collar grip the ram and bear on the cylinder's neck.

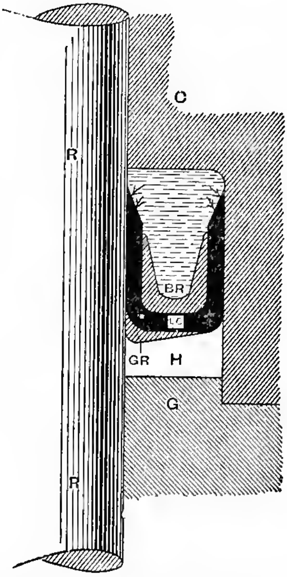
Referring again to the figure of the Bramah press, by taking moments about the fulcrum at F, we obtain the pressure Q, on the plunger of the force pump. Neglecting weight of lever and friction, we get :—

$$P \times AF = Q \times BF.$$

$$\therefore \qquad Q = \frac{P \times AF}{BF}.$$

Further, we know that the statical pressure Q, is transmitted with undiminished force to every corresponding area of the cross section of the ram. Hence,

Q : W : : area of plunger : area of ram.



Leather Collar for a Large Hydraulic Press.

$$\therefore \qquad W \times \text{area of plunger} = Q \times \text{area of ram}.$$

Or, $$W \times \pi r^2 = Q \times \pi R^2.$$

Where r = radius of plunger, and R = radius of ram, both in the same unit. Substituting the previous value for Q, and dividing each side of the equation by π, we get :—

$$W \times r^2 = \frac{P \times AF}{BF} \times R^2.$$

$$\therefore \qquad W = \frac{P \times AF}{BF} \times \frac{R^2}{r^2} = \frac{P \times AF}{BF} \times \frac{D^2}{d^2}.$$

Where D and d are the diameters of the ram and plunger respectively.

Example VII.—In a small Bramah press, P = 50 lbs., A F = 20 ins., B F = 2 ins., area of plunger = 1 sq. in., whilst area of ram = 14 sq. ins. Find W, neglecting friction and weight of lever.

Answer.—By the above formula :—

$$W = \frac{P \times A\,F}{B\,F} \times \frac{R^2}{r^2}.$$

$$\therefore \qquad W = \frac{50 \times 20}{2} \times \frac{14}{1} = 7{,}000 \text{ lbs.}$$

Example VIII.—In Bramah's original press at South Kensington the plunger is 3 ins. in diameter, and it acts at a distance of 6 ins. from the fulcrum, which is at one end of a levor 10 ft. 3 ins. long, carrying a loaded scale-pan at the other end. What should be the pressure of the water in the press in order to lift a weight of 3 cwts. in the scale-pan, neglecting the weight of the lever? Make a diagram of the arrangement. (S. and A. Exam., 1892.)

Answer.—Here $d = 3$ ins., consequently the area of the plunger $= \frac{\pi}{4} d^2 = \cdot 7854 \times 3 \times 3 = 7$ sq. ins., and B F = 6 ins.; A F = 10 ft. 3 ins. = 123 ins.; P = 3 cwts. = 3 × 112 = 336 lbs. Now we have to find the pressure per sq. in. on the ram that will balance P, acting with the stated advantage, since the area of the ram is not given.

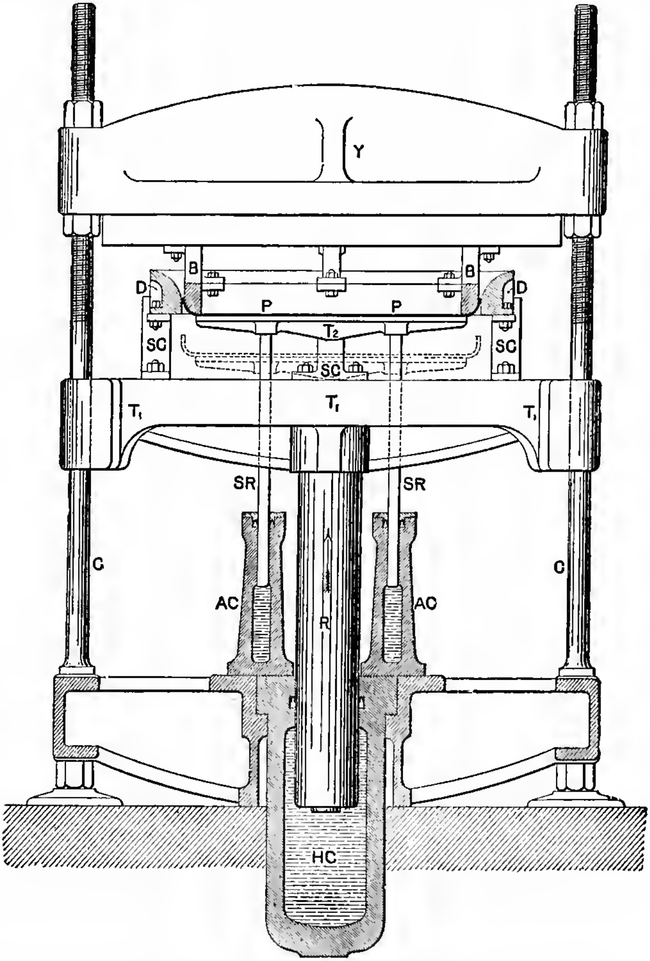
By the above formula :—

$$W = \frac{P \times A\,F}{B\,F} \times \frac{\text{area of 1 sq. in.}}{\text{area of plunger}} = \frac{336 \times 123}{6} \times \frac{1}{7}$$

Or, W = 984 lbs. per sq. in.

Hydraulic Flanging Press.—As an example of the practical application of the Bramah press to modern boiler-making, the accompanying illustration shows the form which it takes when used for flanging. It is worked by a high-pressure water supply derived from a central accumulator, which may at the same time be used to work cranes, punching, riveting, and other similar machine tools.

The operation of flanging the end tube-plates of a locomotive boiler is carried out in the following manner :—The ram R is lowered to near the bottom of the hydraulic cylinder H C, in order to leave room to place the heated boiler plate on the movable table T_2. High-pressure water is then admitted from the central accumulator to the auxiliary cylinders A C, thus forcing the side rams S R, with their table T_2, and the plate P vertically upwards, until the upper surface of the plate bears hard against the bearers B, or internal part of the dies. Water from the same source is now admitted into the hydraulic cylinder H C, and forces up the ram R with its table T_1, supporting columns S C, and the external



Large Hydraulic Press for Flanging Boiler Plates.*

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* The above figure is a reduced copy of one from Prof. Henry Robinson's book on *Hydraulic Machinery*, published by Messrs. Charles Griffin & Co.

part of the disc D, until the latter has quietly and smoothly bent the heated edge of the plate round the curved corner of the internal bearer B. The ram R is now lowered, carrying with it the table T_1 and dies D, by letting out the water from H C. Then the table T_2 with the flanged plate is lowered by letting out water from A C. The plate is removed from its table, allowed to cool, faced, placed in position in the barrel of the boiler, marked off for the rivet holes, drilled, rimed, and riveted in the usual manner. The student will thus understand what a useful and powerful tool a hydraulic press is to the engineer in the hands of a skilful workman; for, it can be made to do better work in far less time, and with far greater certainty, uniformity, and exactitude, than the boiler-smith can turn out, with any number of hammermen to help him. It is fast replacing the steam-hammer for cressing work, and also steam or belt-driven punching and riveting machines, steam cranes, screw and wheel-gear hoists, as well as the screw press for making up bales of goods.

Hydraulic Jack.—This is a combined force pump and hydraulic press arranged in such a compact form as to be readily portable, and applied to lifting heavy weights through short distances. It therefore effects the same objects as the screw-jack, but with less manual effort and with greater mechanical advantage.

The base on which the jack rests is continued upwards in the form of a cylindrical plunger, so as to constitute the ram of the hydraulic cylinder H C. Along one side of this ram there is cut a grooved parallel guide slot G S, into which fits a steel set pin, screwed through the centre of a nipple cast on the side of the cylinder (not shown in the drawings) for the purpose of guiding the latter up and down without allowing it to turn round. The top of the ram has bolted to it a water-tight cup leather C L, by means of a large washer and screw-bolt.

The action of this cup leather is precisely the same as the leather collar in the cylinder of the Bramah press already described; but it has only to be pressed by the water in one direction —viz., against the sides of the truly-bored cast-steel cylinder, instead of against both the ram and the cylinder neck, as in the former case. The head H and upper portion of the machine is of square section, and is screwed on to the hydraulic cylinder in the manner shown by the figure. It contains a water reservoir W R, which may be filled or emptied through a small hole by taking out the screw plug S P.* In the centre line of the head-

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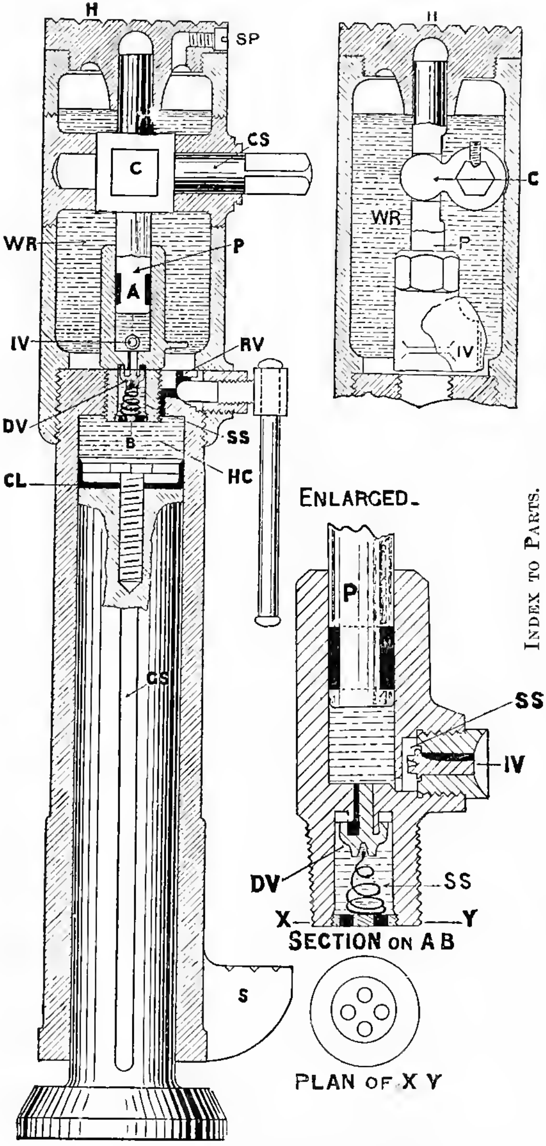
* This screw plug S P is slackened back a little to let the air in or out of the top of the water reservoir when working the jack. There is generally another and separate screw plug opening for filling or emptying the water reservoir, quite independent of the above-mentioned one, which is used in this case for both purposes.



THE HYDRAULIC JACK.

INDEX TO PARTS.

H represents Head for placing under a load.
SP ,, Screw plug for filling and emptying [W R.
W R ,, Water reservoir.
C ,, Crank.
C S ,, Crank shaft.
P ,, Pump plunger.
I V ,, Inlet valve.

D V represents Delivery valve.
S S ,, Spiral springs for I V and D V.
R V ,, Relieve valve.
H C ,, Hydraulic cylinder.
C L ,, Cup leather-packing.
G S ,, Guide slot along the length of arm.
S ,, Step for placing under a load.

piece there is placed a small force pump, the lower end of which is screwed into the centre of the upper end of the hydraulic cylinder. This pump is worked by the up-and-down movement of a handle placed on the squared outstanding end of the turned crank shaft C S. To the centre of the crank shaft there is fixed a crank C, which gears with a slot in the force-pump plunger P, and thus the motion of the handle is communicated to the pump plunger. By comparing the right-hand section of the water reservoir, and the section on the line A B, with the vertical left-hand section of the jack, it will be seen where the inlet and delivery valves I V and D V are situated. On raising the pump plunger P, water is drawn from W R into the lower end of the pump barrel through I V, and on depressing the plunger this water is forced through the delivery valve D V, into the hydraulic cylinder, thus causing a pressure between the upper ends of the cylinder and the ram, and thereby forcing the cylinder, with its grooved head H, and footstep S, upwards, and elevating whatever load may have been placed thereon. Both the inlet and outlet valves are of the kind known as "mitre valves." They have a chamfer cut on one or more parts of their turned spindles, so as to let the water in and out along these channels. The valves are assisted in their closing action by small spiral springs S S, bearing in small cups or hollow centres, as shown more clearly in the case of D V by the enlarged section on A B.

When it is desired to lower the jack, the relief valve R V is screwed back and the water is thus allowed to be forced up again into W R.

Example IX.—Mr. Croydon Marks, in his book on *Hydraulic Machinery*, illustrates and describes another method of lowering the jack-head (first introduced by Mr. Butters, of the Royal Arsenal, Woolwich), where, by a particular arrangement, the inlet and delivery valves are acted upon by an extra depression of the handle, and consequent movement of the pump plunger. He also gives the main dimensions, with a drawing, of the standard 4-ton pattern as used by the British Government, where the ram has a diameter $D = 2$ ins., the pump plunger a diameter $d = 1$ in.; and the ratio of the leverage of the handle to the crank is 16 to 1. Therefore, from the previous formula we find that :—

$$\textit{The Theoretical Advantage} = \frac{W}{P} = \frac{A\,F}{B\,F} \times \frac{D^2}{d^2} = \frac{16}{1} \times \frac{2^2}{1^2} = \frac{64}{1}.$$

And he instances two trials by Mr. W. Anderson, the Inspector-General of Ordnance Factories, to determine the efficiency of these jacks, where, with a pressure on the end of the working handle of

76 lbs., the theoretical load should have been 76 lbs. × theoretical advantage = 76 × 64 = 4,864 lbs., instead of which it was only 3,738 lbs. :—

$$\therefore \qquad 4{,}864 \text{ lbs.} : 3{,}738 \text{ lbs.} : 100 : x.$$

$$\text{Or,} \qquad x = \frac{3{,}738 \times 100}{4{,}864} = 77 \text{ per cent. efficiency.}$$

In a second trial, a load of 1,064 lbs. required a pressure of 22 lbs. on the handle, and consequently the efficiency at this lighter load, as might be expected, was less, or only 74 per cent.

Example X.—With a hydraulic jack of the dimensions given above, and of 77 per cent. efficiency, it is desired to lift a load of 4 tons; what force must be applied to the lever handle?

Answer.—By the previous theoretical formula :—

$$W = \frac{P \times A\,F}{B\,F} \times \frac{D^2}{d^2}$$

$$\therefore \qquad P = \frac{W \times B\,F}{A\,F} \times \frac{d^2}{D^2}$$

$$,, \quad = \frac{4 \times 2{,}240 \times 1}{16} \times \frac{1^2}{2^2} = 140 \text{ lbs.}$$

But the efficiency of the machine is only 77 per cent., consequently 140 lbs. is 77 per cent. of the force required :—

$$\therefore \qquad 77 : 100 :: 140 \text{ lbs.} : x \text{ lbs.}$$

$$x = \frac{140 \times 100}{77} = 181{\cdot}81 \text{ lbs.}$$

Example XI.—Show, with the aid of sectional sketches, the construction of the ordinary hydraulic lifting jack. If, in such a machine, the mechanical advantage of the lever or handle is 12 to 1, and the diameter of the lifting ram is 2 inches, while the diameter of the plunger is $\frac{7}{8}$ of an inch, what weight can be lifted theoretically when a pressure of 50 lbs. is applied to the lever handle? (S. & A. Exam., 1891.)

Answer.—The hydraulic jack has been fully described and illustrated in this lecture.

Let D = Diameter of ram = 2 inches.
,, d = ,, plunger = $\frac{7}{8}$ inch.
,, n = Mechanical advantage of lever = 12 : 1.
,, P = Effort applied at end of lever = 50 lbs.
,, W = Weight raised.

If Q denotes the pressure on the plunger, caused by the effort P applied at the end of the lever, then :—

$$Q = P n.$$

But, $$\frac{W}{Q} = \frac{\text{area of ram}}{\text{area of plunger}} = \frac{D^2}{d^2}.$$

$$\therefore \quad \frac{W}{Pn} = \frac{D^2}{d^2}.$$

This is a formula giving the relation between W, P, n, D, and d, which is also true for the hydraulic press.

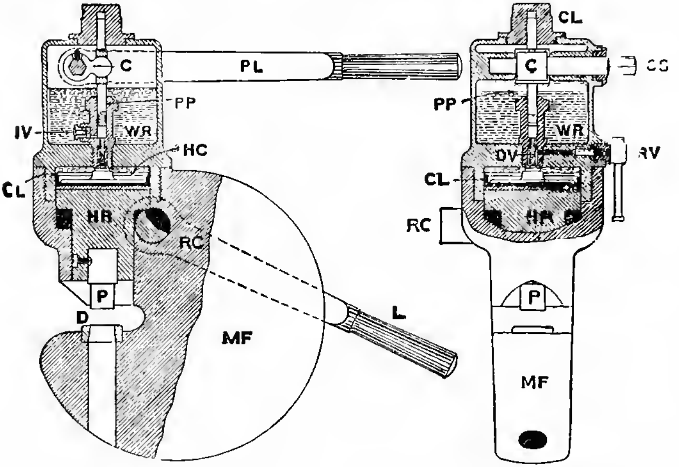
Substituting the above values in this formula, we get :—

$$\frac{W}{50 \times 12} = \frac{2^2}{(\frac{7}{8})^2}.$$

$$\therefore \quad W = 3{,}134 \text{ lbs., } or = 1{\cdot}4 \text{ tons, nearly.}$$

Hydraulic Bear.—This is another very useful application of the hydraulic press and force pump. It is used in every shipbuilding-yard and bridge-building works. By comparing the drawing with the index to parts, it may be seen that its construction and action are similar to the hydraulic jack just described in full detail, and we need say nothing more than direct the student's attention to the action of the raising cam, and to the means by which the apparatus is lifted and suspended. In order to raise the punch P, for the admittance of a plate between it and the die D, the relief valve R V, must first be turned backwards, and the lever L, depressed. This causes the corner of the raising cam R C, to force the hydraulic ram H R, upwards, and the water from the hydraulic cylinder H C, back into the water reservoir W R. The relief valve R V, may now be closed and the plate adjusted in position. Then the pump lever P L, can be worked up and down until the punch P, is forced through the plate, and the punching drops through the die hole D, in the metal frame M F, to the ground, or into a pail placed beneath to receive it.

The whole bear is suspended by a chain (worked by a crane or other form of lifting tackle) attached to a shackle, whose bolt passes through a cross hole in the back of the metal frame M F, just above, but a little to the front of, the centre of gravity of the machine. This hole and shackle are not shown in the drawing, but the student can easily understand that the hole would be bored a little above where the letters R C, appear on the side view, and that the chain would pass clear of the pump lever, since this works well to the right-hand side of the bear.



Side View and Section. End View and Section.

The Hydraulic Bear, or Portable Punching Machine.

Index to Parts.

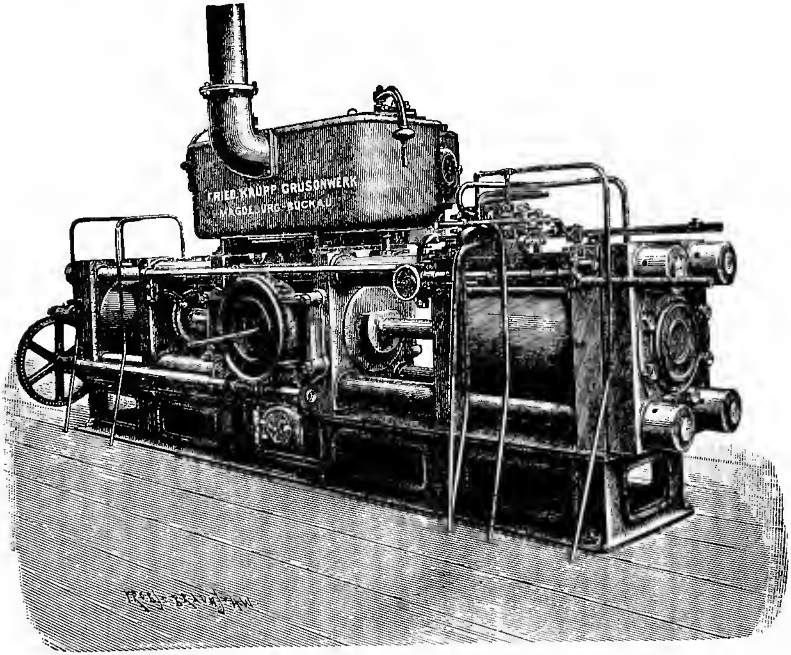
P L	represents	Pump lever.	H C	represents	Hydraulic cylinder.
C S	,,	Crank shaft.	C L	,,	Cup leather.
C	,,	Crank.	H R	,,	Hydraulic ram.
P P	,,	Pump plunger.	R C	,,	Raising cam.
W R	,,	Water reservoir.	L	,,	Lever for R C.
I V	,,	Inlet valve.	P	,,	Punch.
D V	,,	Delivery valve.	D	,,	Die ring.
R V	,,	Relief valve.	M F	,,	Metal frame.

Lead-covering Cable Press.—Hydraulic presses are now employed for making lead pipes, and for covering electric light cables with a close-fitting tube of lead. For the former purpose the lead is heated until it is nearly melted in a strong chamber from which it is forced by rams to squirt through a die at the top of the machine. A mandrel projects into the centre of the die, and, consequently, the lead issues as a continuous tube.

The accompanying figure illustrates a press for covering electric cables with lead in this manner, as carried out by Messrs. Siemens Brothers, at their Woolwich works. It consists essentially of a receiver in which two rams work, and which contains a mandrel and matrice. The lead is melted in the melting pot at the top by gas or petroleum, and is then poured into the receiver below.

The cable enters at one side of the receiver, and passes through the mandrel and leaves at the other, while as soon as the rams begin to force their way into the receiver, the lead casing is formed round the cable.

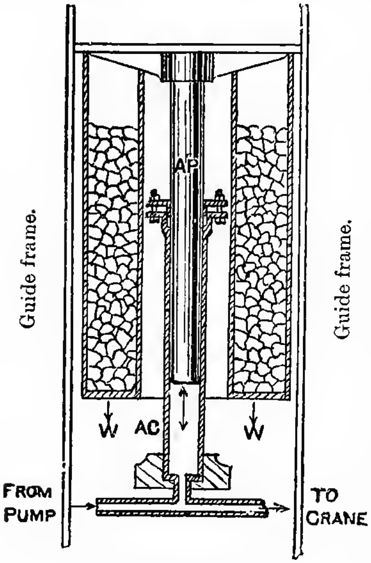
The matrice can be so nicely adjusted by means of steel cones, that a lead casing of perfectly uniform thickness of a fraction of a millimetre, can be obtained. An excellent feature of this press is, that it possesses two rams which, by distributing the pressure more uniformly, ensure a more regular casing than is possible with only one, as is usually the case. The two rams are connected together by an adjusting apparatus so that they will both move forward at the same rate. The mandrels and matrices can easily be changed to suit cables whose diameters range from $\frac{1}{8}$ in. to $2\frac{3}{4}$ ins.



PRESS FOR COVERING CABLES WITH LEAD, BY FRIED. KRUPP GRUSONWERK, GERMANY.

Hydraulic Accumulator.—The demand for hydraulic power to work elevators, cranes, swing bridges, dock gates, presses, punching and riveting machines, &c., being of an intermittent nature—at one moment requiring a full water supply at the maximum pressure, and at another a medium quantity, whilst in many cases all the machines may be idle—it is evident that if an engine with pumps were devoted to supplying this demand in a direct manner, the power thereof would have to be equal to the greatest requirements of the plant, and would have to instantly answer any and every call from the same. In the case of a low-pressure supply, as for lifts, this difficulty is best overcome by placing one tank in an elevated position at the top of the hotel or building where the lift is required, and another tank below the level of the lowest flat. Then a small gas engine working a two or three-throw pump, or a Worthington duplex steam pump, may be used to elevate the water more or less continuously from the lower to the higher tank. The "head" of water in the elevated tank will, if sufficient, work the lift at the required speed, and the discharged water from the hydraulic cylinder will enter the lower tank, to be again sent round on the same cycle of operations. Should the lift be stopped for any considerable time, then a float in the upper tank, connected by a rope or chain with the shifting fork for the belt-driven pumps (in the case of the gas engine) will force the belt over on to the loose pulley, or shut off the steam from the Worthington pump. And when the water falls in the upper tank, the float will cause a reverse movement of the rope and shift the belt to the tight pulley, or open the steam valve, and so start the pumps.

When the pressures required are great, such as for cranes, &c., where 700 lbs. on the square inch is considered a very medium pressure, an elevated tank would be out of the question, for it would have to be fully 1600 feet high in order to exert this force and to overcome friction. Under these circumstances recourse is had to a very simple and compact arrangement called an accumulator, of which we here give a lecture diagram, without any details of cocks or valves, and automatic stopping and starting gear. A steam engine, or other motor, works a continuous delivery pump, of the combined piston and plunger type, without an air vessel, as already illustrated in this lecture. The water from the pump enters the left-hand branch pipe leading into the foot of the accumulator cylinder, and forces up the accumulator ram with its crosshead or top T-piece, and the attached weight or dead load, until the ram has reached nearly to the end of its stroke. Then the top of the T-piece, or a projecting bracket on the side of the wrought-iron cylinder containing the dead load, engages with and lifts a small weight attached to a chain passing over a pulley fixed to the guide frame or to the wall of the accumulator house. This chain is connected directly to the throttle valve of the steam supply pipe, or to the belt-shifting gear if the pump is driven by belt gearing, and being provided with a counter-weight, the motor and pump are automatically stopped by the raising of the weight and the chain in the accumulator house. Should the water which has been forced into the accumulator cylinder be now used by a crane or other machine, the load on the ram causes it to follow up and keep a constant pressure on the water. The starting weight falls as the receding T-piece or bracket descends, and thus pulls the starting chain, and opens the steam engine throttle valve, or shifts the belt from the loose to the fixed pulley, and again sets the pump to work. Should the hydraulic machines be working continuously, then the pump is kept going, for the water from it passes directly on to the machines, and only the surplus water finds its way into the accumulator cylinder if the pump's supply exceeds the demand of the machines for water.



The Hydraulic Accumulator.

Index to Parts.

A C for Accumulator cylinder.
A P ,, Accumulator plunger or ram.
W ,, Weight or load contained in an annular cylinder of wrought iron and suspended from the top of T-piece or crosshead.

The annular cylinder of wrought iron is generally filled with scrap iron, iron slag, sand, or other inexpensive heavy material. The accumulator cylinder A C, has a stuffing-box and gland at its upper end. A coil of hemp woven into a firm rectangular section and smeared with white lead is placed in the bottom of the stuffing-box. The gland is screwed down on the top of this packing until at the normal pressure the water in the cylinder cannot leak past it. Cup leather packing is seldom used for this simple form of accumulator; just the ordinary packing that would be used for pump rods is found to answer all requirements. This is the simplest form of accumulator which we have here described, but other forms will be illustrated in the next lecture.

EXAMPLE XII.—Describe and sketch in section an hydraulic accumulator, showing how the ram is kept tight in the cylinder. An hydraulic press, having a ram 16 inches in diameter, is in connection with an accumulator which has a ram 8 inches in diameter and is loaded with 50 tons of ballast; what is the total pressure on the ram of the press? (S. & A. Exam., 1892.)

ANSWER.—The first part of the question is answered by the previous figure and by the text.

By Pascal's Law the pressure *per square inch* in the accumulator is equal to the *pressure per square inch in the* hydraulic press. Consequently :—

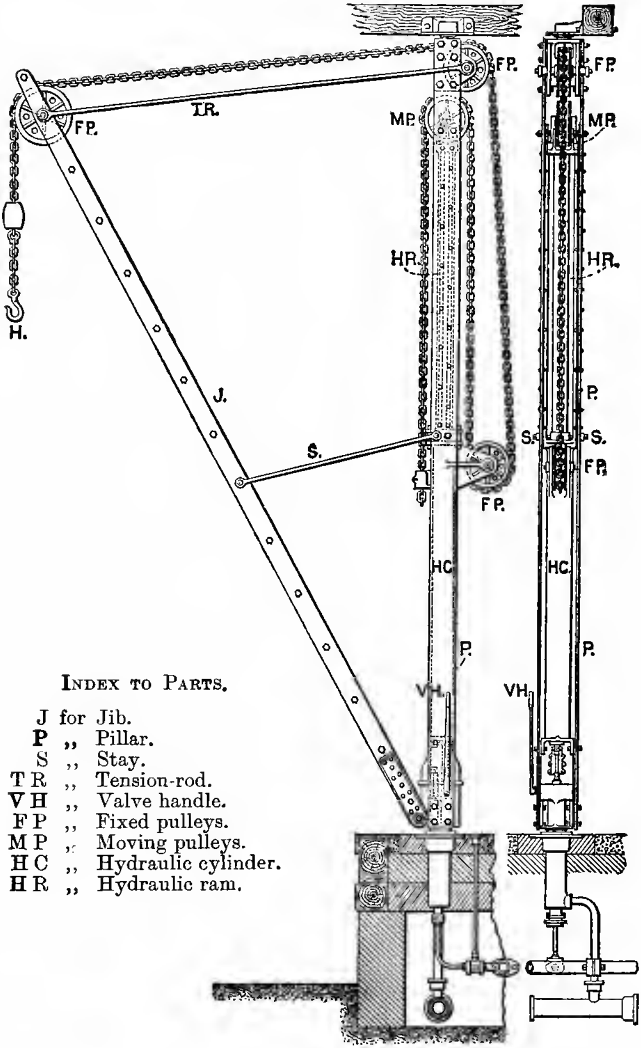
$$\frac{\textit{Total Pressure on Press}}{\textit{Total Load on Accumulator}} = \frac{\textit{Cross Area of Press Ram}}{\textit{Cross Area of Accumulator Ram}}$$

$$\frac{P}{50} = \frac{\pi}{4} \times 16^2 \div \left(\frac{\pi}{4} \times 8^2\right) = \frac{16^2}{8^2}.$$

$$\therefore \qquad P = \frac{50 \times 16 \times 16}{8 \times 8} = 200 \text{ tons.}$$

Hydraulic Cranes.—Another very common application of hydraulic power is the working of cranes for handling goods at wharves, warehouses, or railway depôts, and we illustrate a simple one for the latter purpose. In this crane, the lower part of the pillar forms the cylinder for a ram H R, which carries a pulley M P, at its upper end and is actuated by the water pressure. A chain, fastened at one end to the crane pillar, passes over this pulley and then round two fixed ones F P, before going to the jib pulley. Hence, when the ram is forced up, the chain will be pulled up twice as far. A small slide valve, actuated by the handle V H, controls the supply of water to the hydraulic cylinder. By means of this valve, the cylinder is put into communication with either the pressure or the exhaust pipe, or it may be cut off from both. The weight of the ram and load attached to the hook H, are utilised to drive out the water on the return stroke.

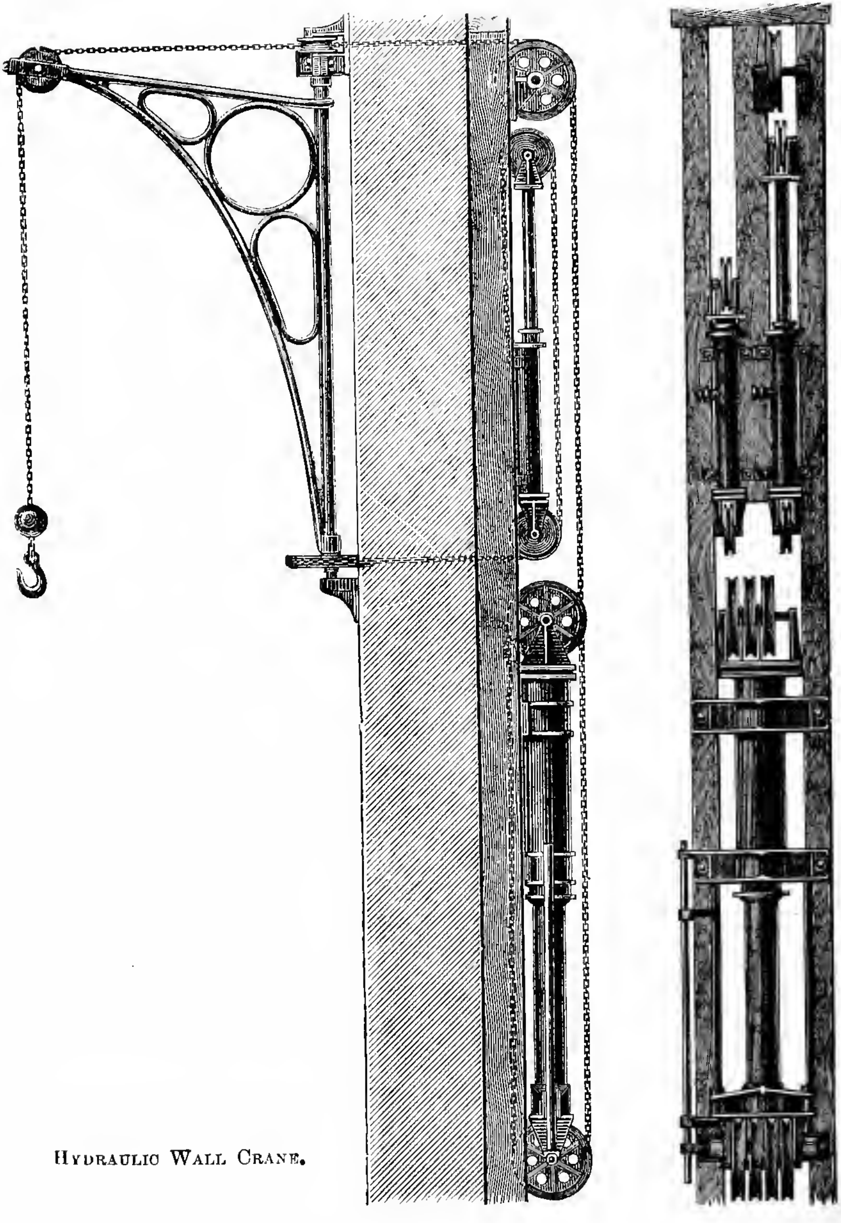
Hydraulic Wall Crane.—Our next figure shows a crane on the same principle fixed to the wall of a warehouse or shed. In this case, however, the motion of the ram is magnified eight times by placing four pulleys side by side on the end of the large ram which here moves downwards. The slewing of this crane is also accom-



INDEX TO PARTS.

J for Jib.
P ,, Pillar.
S ,, Stay.
T R ,, Tension-rod.
V H ,, Valve handle.
F P ,, Fixed pulleys.
M P ,, Moving pulleys.
H C ,, Hydraulic cylinder.
H R ,, Hydraulic ram.

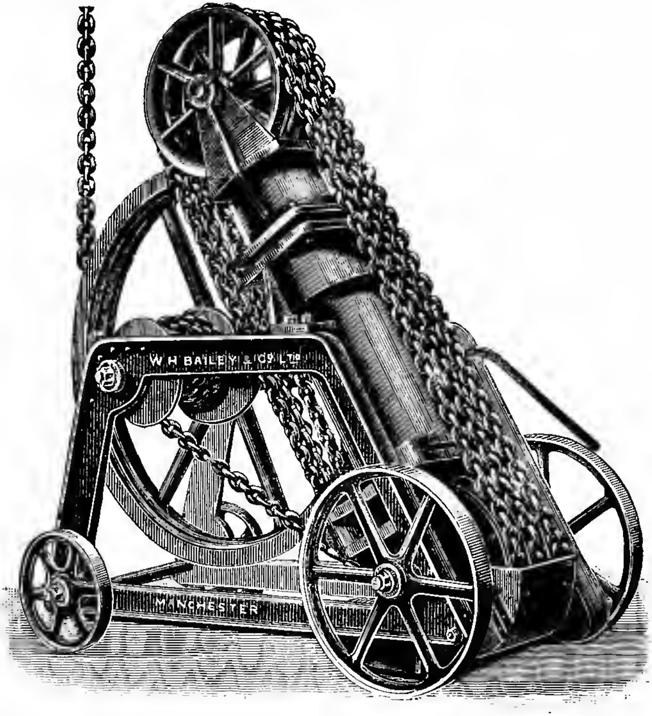
HYDRAULIC CRANE.



HYDRAULIC WALL CRANE.

plished by hydraulic means as follows:—Each end of a chain, which passes round a wheel at the bottom of the crane, is taken over a pulley on a separate small ram, and then fixed to the framework. Matters are so arranged, that when these two smaller rams are at their half strokes the crane projects at right angles to the wall. If water be admitted below one of the rams that end of the chain is forced up and the crane is hauled round. At the same time, the other ram is pulled down by the chain so as to be ready to bring the crane back when required.

Movable Jigger Hoist.—For lifting light loads, say under a ton,


W H BAILEY & Cº LTD
MANCHESTER

MOVABLE JIGGER HOIST BY W. H. BAILEY & CO., LTD., MANCHESTER.

from a ship's hold, jigger hoists are often employed. The lifting chain or rope is wound on the large drum, while the ram actuates another chain coiled on a small drum on the same axis. Conse-

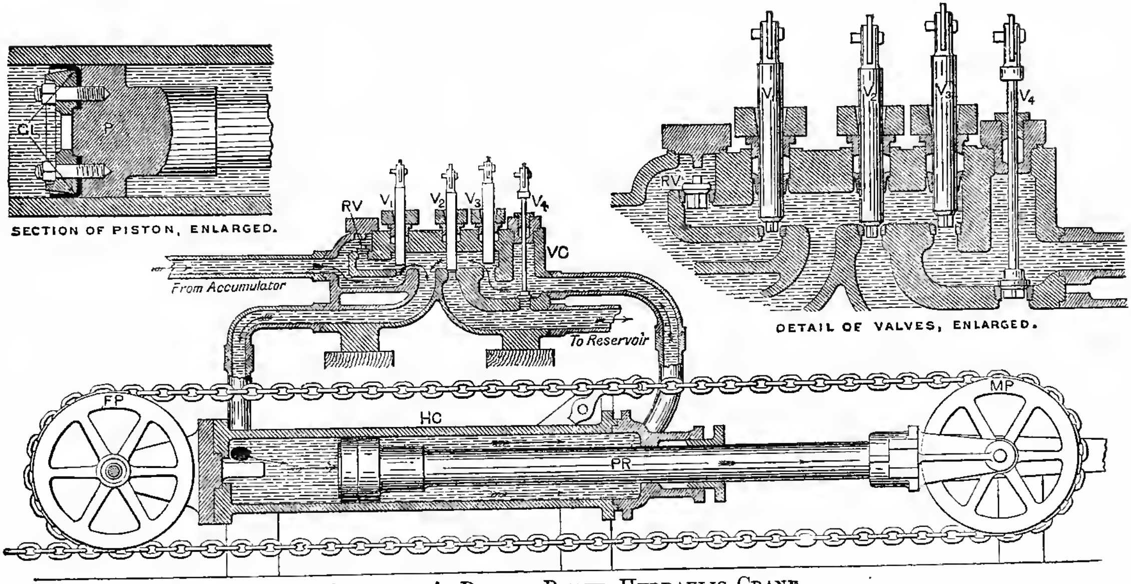
4

quently, the load is lifted very rapidly, as the motion is magnified first by the sheaves on the ends of the ram and cylinder, and then by the wheel and axle arrangement. The jigger stands on the quay and the lifting chain passes over a pulley hung from one of the ship's derricks. The valve gear can be worked, if desired, by a rope from the ship's deck.

Double Power Hydraulic Crane.—A disadvantage of hydraulic pressure apparatus is, that, as a rule, the same amount of water is used with a light load as with a heavy one, any surplus energy being consumed in fluid friction. To reduce this loss Lord Armstrong employed a combined ram and piston as shown in the figure on next page.

With the valves in the position shown, both ends of the cylinder are in communication with the supply pipe, and the ram is moved forward by the pressure on the difference between the areas of the piston and ram. The water used also corresponds to this difference of area because that in the right-hand end of the cylinder is forced round to the left, and only the remainder comes from the accumulator. Now, if we require to lift a greater load than the crane will raise under these conditions, V_3 must be closed and V_4 opened. This allows the water at the front of the piston to escape, and we get the full pressure on the back of the piston. To lower the load, V_1 is closed and V_2 opened. Then, the water escapes from the left of the piston partly to the exhaust and partly to the other end of the cylinder. The relief valve allows a little water to escape back to the supply pipe whenever the pressure in the valve chamber rises above that of the accumulator, owing to the sudden closing of the outlet valve. This gear is designed to work horizontally in a chamber sunk into the ground below the crane.

Hydraulic Capstan.—When a continuous rotary motion is required from hydraulic power, the usual method is to put three single-acting oscillating cylinders in symmetrical positions round the shaft, with all three pistons acting on one crank pin, while the motion of the cylinder itself uncovers the openings for the inflow and exit of the water at the proper time. As an example, we illustrate a hydraulic capstan for use on dock quays. In this one there are four cylinders with their mid positions 90° apart. A plunger in each acts on the crank below the capstan. The water enters and leaves by passages in the trunnions on which the cylinders swing, and these passages are opened at the right moment by the oscillation of the cylinder



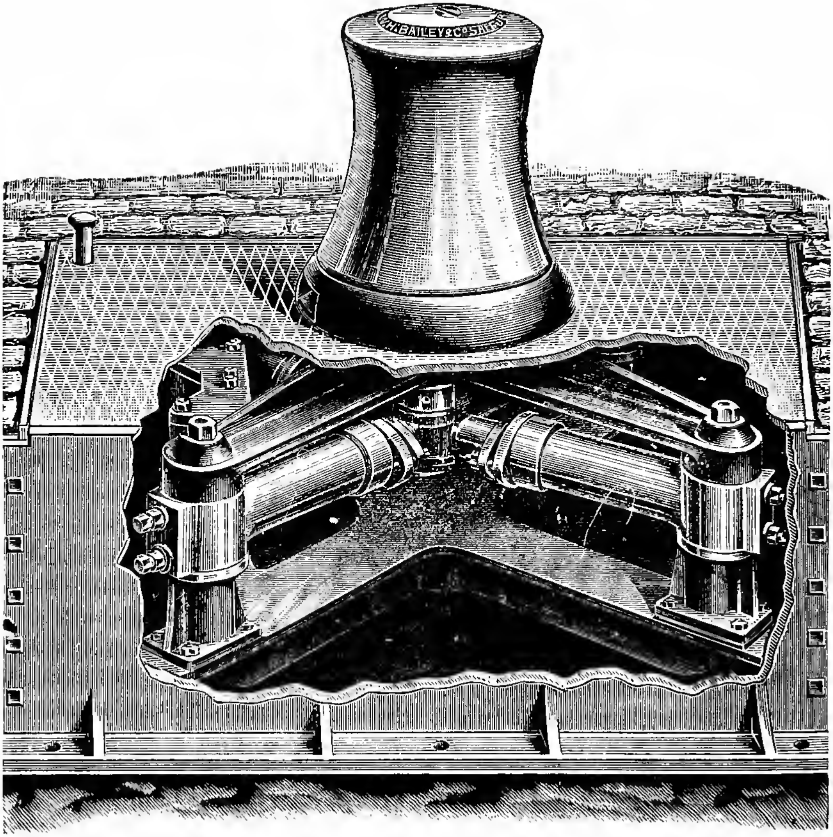
SECTION OF PISTON, ENLARGED.

DETAIL OF VALVES, ENLARGED.

Armstrong's Double Power Hydraulic Crane.

H C for Hydraulic cylinder.
P R ,, Piston-rod.
P ,, Piston.
C L ,, Cup leather.
V C ,, Valve chamber.

V_1 for Inlet valve.
V_2 V_4 ,, Outlet valves.
V_3 ,, Connecting valve.
R V ,, Relief valve.



Hydraulic Capstan by W. H. Bailey & Co., Ltd., Manchester

Lecture I.—Questions.

1. One side of a reservoir has a slope of 12 vertical to 5 horizontal; what is the whole amount of the pressure of the water against 50 feet of its length, when the depth of the water is 12 feet? *Ans.* 243,750 lbs.

2. In an empty dock the water is level with the sill at the lowermost edge of the dock gate, the level of the water on the opposite side of the gate being 10 feet above the sill. The dock gate is 10 feet wide, find the pressure in pounds on the dock gate. If the water were at a level of 5 feet in the dock, the level outside being the same as before, how much would the pressure on the gate be relieved? *Ans.* 31,250 lbs.; 7,812·5 lbs.

3. State Archimedes' principle. A cylindrical can is 6 inches in diameter and 30 inches deep, and is required, when empty, to stand in a bath of water 30 inches deep without being lifted up. To what weight must the can be loaded, the weight of a cubic foot of water being 62½ lbs.? *Ans.* 30·679 lbs.

4. A rectangular tank, 4 feet square, is filled with water to a height of 3 feet. A rectangular block of wood, weighing 125 lbs., and having a sectional area of 4 square feet, is placed in the tank, and floats with its sides vertical and with this section horizontal. How much does the water rise in the tank, and what is now the pressure on one vertical side of the tank? *Ans.* 1½ inches; 1,220·6 lbs.

5. The total force in the direction of the motion of a piston is the cross-sectional area of the cylinder multiplied by the pressure. Why is this so, the piston not having a plane surface?

6. An escape valve, loaded partly by a weight and partly by a spring, is fitted to a main conveying water under pressure, and is required to open automatically when the water pressure rises above a certain amount. Sketch and describe the construction of such a valve when arranged on the double beat principle, and explain clearly the hydrostatic principle involved therein.

7. Prove that when a thin spherical shell is exposed to the bursting pressure of gas or liquid the stress in the material is half as great as that within the curved surface of a thin cylindrical shell exposed to the like pressure, each shell being of the same thickness and diameter.

8. Sketch a combined plunger and bucket pump with index of parts, explaining its use and action, also sketch its application to the lifting of water from deep mines. Suppose a pump raises 5,000 gallons every half-minute from a depth of 600 feet with 30 per cent. loss in system, what is the horse-power required? *Ans.* 2,597 H.P.

9. Describe a force pump for supplying water to the accumulator of hydraulic cranes. Sketch a section through the plunger and valves.

10. Sketch and describe a force pump having a solid plunger, showing the construction of the valves. The diameter of the plunger is 2½ inches, and it is driven by a crank 2 inches in length making 30 revolutions per minute. Find the cubic inches of water pumped in 5 minutes. *Ans.* 2,945 cubic inches.

11. Describe and illustrate by a longitudinal section, and such other views as may be necessary, the construction and action of a double-acting pump

and its valves, supposing the pump cylinder to be of 3½ inches internal diameter, and to work at a pressure of 700 lbs. per square inch. Of what materials would the several parts be constructed?

12. A vertical single-acting pump has to elevate water 50 fathoms. The bore of the pump is 6 inches; stroke, 6 feet; number of up strokes, 10 per minute. Find (*a*) the pressure per square inch on the pump bucket when it is at the bottom of its stroke; (*b*) the weight of water discharged per minute; (*c*) the horse-power of the engine required to drive the pump, supposing 30 per cent. of the engine-power to be lost by friction, &c. Sketch an arrangement of the kind. *Ans.* (*a*) 130·28 lbs. per square inch; (*b*) 736·31 lbs.; (*c*) 9·56 H.P.

13. Give any method with which you are acquainted whereby the valves in a pumping engine may be relieved from the shock due to the inertia of the water in the mains.

14. A pump for exhausting air from a receiver has a solid piston and one valve in the casing. Describe, with a sketch, the construction of the pump, and explain the nature of the improved form known as Sprengel's pump, where a small portion of mercury forms the piston, and no valve is required.

15. Describe and show, with the aid of necessary sketches, the construction of the "Pulsometer." Describe how it works, and indicate the contrivances introduced to promote the steady flow of water and to prevent sudden shocks upon the apparatus. Is the pulsometer an economical arrangement for raising water? Give reasons for your answer. What, if any, are its advantages over the ordinary piston pump?

16. Sketch in section the cylinder, ram, and leather collar of an hydraulic press. Explain the principle of the press and the manner in which the escape of water is prevented. Example—The sectional area of the plunger of the force pump is $\frac{1}{50}$ that of the ram, and the leverage gained by the pump handle is 12 to 1, find the pressure on the ram when a force of 60 lbs. is exerted at the end of the pump handle. *Ans.* 36,000 lbs.

17. Describe, with a sectional sketch, a hydraulic press where the ram is actuated in both directions. Show the position and forms of the cup leathers.

18. The return stroke in an hydraulic press is often accomplished by forming the ram like a piston with a very large piston-rod. Sketch in longitudinal section such a press, showing the arrangement of the leathers. What will be the relative speeds of the forward and return strokes of the ram when the larger and smaller diameters are 15 inches and 14 inches respectively, the pumps for the supply of water running at the same speed in both cases?

19. In some hydraulic presses a single valve, held down by a lever and weight, is used both to indicate and relieve the pressure. Sketch the valve in position and explain its action.

20. Describe clearly and show with sketches the construction and action of any one form of portable hydraulic riveting machine with which you are acquainted. Show clearly the valves and connections by which the pressure is applied to close the rivet, and how the pressure is released and the tool withdrawn from the rivet head when the riveting is completed. How is the water pressure conveyed to the machine?

21. What are the advantages in forging large masses of steel by hydraulic pressure over the same operation performed by the steam hammer? Show clearly, with the assistance of the necessary sketches, the method em-

ployed in hydraulic forging presses for bringing the ram or pressing surface rapidly back from the work after each application of the pressure. (S. & A. Hons. Exam., 1896.)

22. Sketch a section through an hydraulic lifting jack and hydraulic bear, and describe the manner in which the pressure exerted on the handle is transmitted to the ram.

23. A 4-ton hydraulic lifting jack has a lifting ram of 2 inches in diameter, and a pump plunger of 1 inch in diameter. The jack is worked by a lever handle, the leverage being 16 to 1. What pressure must be applied at the end of the handle in order to lift a load of 25 cwts., if the efficiency of the machine is 80 per cent.? Make a vertical section of the jack, showing the valves and the mode of connecting the lever with the pump plunger. How can the weight be lowered slowly and regularly without jerks? *Ans.* 54·37 lbs.

24. Explain, with the aid of a sectional sketch, the action and construction of the hydraulic jack. How is the pressure taken off and the load slowly lowered? If the ram is 2 inches in diameter, the pump plunger $\frac{7}{8}$ inch, and the mechanical advantage of the handle 10, what is the total mechanical advantage, neglecting friction? (S. & A. Adv. Exam., 1897.) *Ans.* 52·26 : 1.

25. Make a sketch of a 10-ton hydraulic jib crane in which the lifting cylinder is carried in the pillar or post of the crane. What would be the diameter of ram required in the arrangement you adopt, supposing water to be supplied to the crane at a pressure of 700 lbs. per square inch, neglecting friction, &c.? *Ans.* Diameter = 6·37 inches.

26. What is the object of a relief valve in an hydraulic crane, and where is it placed? Sketch a longitudinal section through the cylinder and ram of a crane working at two powers. Explain the mode of action.

27. Describe, with sketches, some form of hydraulic lift, and the manner in which it is worked.

28. Describe with sketches a direct-acting 80 foot hydraulic lift, to carry a load of 1 ton, the ram is 4 inches diameter, and the moving parts are balanced by a counterweight and chain. The lift is worked from a tank of water 40 feet above the highest level of the platform and an intensifier must be used. (S. & A. Hons. Exam., Part II., 1898.)

29. Make a section through the cylinder of an hydraulic crane with two powers as applied for raising weights. Give a general description of the crane, and explain the mode of working it by referring to your sketch.

30. Describe the general construction of the lifting apparatus in the hydraulic crane, making a sketch of the cylinder and ram. Show the method of obtaining *two* powers from a single cylinder. In what way is the *pulley principle* applied in such cranes?

31. In an hydraulic crane with two powers show (1) the apparatus for lifting the weight, (2) the method of slewing or rotating the jib of the crane.

32. Explain the advantage to be derived from making the length of stroke of an hydraulic engine adjustable. Describe and give sketches of a construction for this purpose.

33. Prove the formula used for finding the resultant pressure on a sloping plane interface, with any shape of boundary underneath the surface of a liquid. Also find the position of the resultant. Apply the formula to the case of a rectangle with its highest horizontal side 5 feet long at 20 feet below the surface, its other sides being 12 feet long and making 30° with the horizontal. (S. and A. H., Part I., 1899.)

34. A rectangle with its highest horizontal side 2 feet long at 10 feet below still water level, its other sides being 3 feet long, making 43° with the horizontal; find the amount and position of the resultant pressure upon the area. Prove the truth of your formulæ. *Ans.* 4,121 lbs.
(B. of E. H., Part I., 1900.)

35. The hydraulic company's pipes are to stand a maximum working pressure of 1,200 lbs. per square inch; internal diameter 4 inches. Find the outside diameter, the working tensile stress in the material being 3,200 lbs. per square inch. When this stress is reached inside, what is the stress outside? *Ans.* Outside diameter 5·93 inches, say 6 inches.
(B. of E. H., Part I., 1900.)

36. Describe, with sketches, a hydraulic lift with hydraulic method of balancing. (B. of E. H., Part I., 1901.)

37. Describe, with sketches, any machine with which you are acquainted, in which water at great pressure is employed. Point out at what places solid friction takes effect. What are the differences in character between solid and fluid friction? If 3 cubic feet of water at 700 lbs. per square inch gauge pressure are employed in a crane to lift a weight of 2 tons through a height of 50 feet, what is the efficiency of the crane?
(B. of E. Adv., 1901.)

38. Describe, with detail sketches, a good form of hydraulic crane, with arrangements for varying the water consumption with the load lifted.
(C. & G., 1900, H., Sec. C.)

39. Describe, with sketches, either (*a*) a hydraulic riveter, or (*b*) a hydraulic boiler flanging press. Obtain an expression for the maximum pressure either can exert. Assume your own factors for friction losses.
(C. & G., 1900, H., Sec. C.)

40. In a hydraulic crane the water pressure employed is 750 lbs. per square inch; to lift the full load of 10 tons three cylinders are used, each 10 inches in diameter, and the load by an arrangement of pulley blocks is lifted at six times the speed at which the rams move out of their cylinders. How many foot-pounds of work are done on the rams in lifting the 10 tons 30 feet, and how many foot-pounds are wasted?
(C. & G., 1901, O., Sec. A.)

41. Explain, with careful sketches, the construction and working of either (*a*) a hydraulic jack, *or* (*b*) a hydraulic compression or baling press. In each case the method of packing the moving parts must be fully described. How are the sizes of the rams determined? What data are needed? (C. & G., 1901, H., Sec. C.)

N.B.—*See Appendices B and C for other questions and answers. This note refers to all the Lectures.*

LECTURE I.—A.M.INST.C.E. EXAM. QUESTIONS.

1. Prove that a body of any form when immersed or floating in a liquid is acted on by a force equal to the weight of the liquid displaced, acting vertically upwards through the centre of gravity of the displaced volume. A bulb weighing 12 oz. is found to weigh 8 oz. when immersed in water, and 7 oz. when immersed in another liquid. What is the specific gravity of the liquid? (I.C.E., *Oct.*, 1897.)

2. Show how to find the resultant thrust due to the pressure of water on a rectangular plane surface, two of the sides of the rectangle being horizontal. A vertical wall, 2 feet thick and 18 feet high, weighing 124 lbs. per cubic foot, supports the pressure of water on one side. How high may the water rise without causing the resultant force on the base of the wall to pass more than 8 inches from the middle of the wall's width? (I.C.E., *Feb.*, 1898.) *Ans.* 6·7 feet.

3. The centre of a spherical body, 6 inches diameter, is 3 feet below the surface of water, what is the resultant pressure on the surface of the body? (I.C.E., *Oct.*, 1898.)

4. A dock entrance, whose level floor is 24 feet below the water, has a width of 62 feet at the water-level and 50 feet at the floor, the side-walls being built with a straight batter. The entrance is closed by a caisson, and on one side of the caisson the floor is dry. Calculate the total horizontal pressure upon the caisson, and the height of its centre of action above the floor. (I.C.E., *Oct.*, 1898.) *Ans.* 1,044,000 lbs.; 8 ft. above the floor.

5. A rectangular area inclined at 20° to the horizontal is submitted to water-pressure. Its side is 11 feet and is horizontal, and is 30 feet below the water-level; its parallel side is 62 feet below the water-level; what is the total pressure on the area counting from atmospheric pressure? (I.C.E., *Oct.*, 1898.)

6. Describe a bucket-and-plunger pump. What causes limit its efficiency? (I.C.E., *Feb.*, 1899.)

7. In a common form of hydraulic crane the piston is 8 inches in diameter; the speed at which the weight is raised is 8 times that of the piston. If the water acting on the piston is at a pressure of 750 lbs. per square inch, find what weight may be lifted by the chain if the efficiency of the mechanism be 55 per cent. What is the chief cause of the low efficiency? How many gallons of water are used in a lift of 30 feet? I.C.E., *Feb.*, 1899.) *Ans.* 2,591·7 lbs.; 8·18 gallons of water.

8. A block, in the form of an isosceles wedge, 1 foot high and having a base 1 foot square, is immersed in water with its base horizontal and uppermost at a depth of 4 feet below the surface. Determine the pressure, and the vertical component of the pressure on each face. Hence show what must be the weight of the wedge so that it may just float. (I.C.E., *Feb.*, 1899.)

9. If a spherical buoy 6 feet in diameter floats in sea-water with half its volume immersed, what is the weight of the buoy? Find also what will be the load draught of a flat-bottomed pontoon of rectangular form, whose length is 175 feet and its width 20 feet, when the weight of pontoon and load amounts to 400 tons. (I.C.E., *Feb.*, 1899.)

10. Suppose that the pontoon last mentioned is to be symmetrically loaded in such a manner as to bring the centre of gravity to a height of 6 feet above the deck; and determine the question whether the pontoon would in that case float upright in calm water, or whether it would capsize. (I.C.E., *Feb.*, 1899.)

11. How high may water rise on one side of a vertical wall 9 feet high, 27 inches thick, of material weighing 143 lbs. per cubic foot, before the resultant force at the base of the wall passes outside the edge? (I.C.E., *Feb.*, 1899.)

12. A square sluice-opening, 3 feet by 3 feet, is formed in the vertical face of a dam, and the upper edge of the opening is 6 feet below water. Calculate the hydrostatic pressure upon the square area, and find the exact position of its centre of action. (I.C.E., *Feb.*, 1899.) *Ans.* 1⅖ feet from the bottom of flap.

13. A vessel with a square base has three vertical sides and one sloping side, so that the shape of the top is a rectangle, with one pair of sides equal to those of the base and the other pair half that amount; the height of vessel is equal to the side of base. If it is filled with water, what is the amount and distribution of the pressure on the base. If the water freezes into a block, how is the distribution of pressure altered. (I.C.E., *Oct.*, 1899.)

14. Sketch a lift-pump, a double-acting force-pump, and a bucket-and-plunger pump, and state generally their relative advantages.
(I.C.E., *Oct.*, 1899.)

15. The sides of a circular tank 15 feet in diameter are constructed of iron plates ½ inch thick, riveted together. Omitting the effect of the joints and the connection with the flat bottom of the tank, find the depth to which it may be filled with petroleum weighing 55 lbs. per cubic foot, so that the maximum stress on the metal may not exceed 1,000 lbs. per square inch. (I.C.E., *Feb.*, 1900.)

16. Give the theory of the simple hydraulic press, and state the chief causes of loss of mechanical advantage in practice. A hydraulic punch has a ram 8 inches diameter; ⅝-inch holes are being punched, each requiring a force of 70,000 lbs. What is the water pressure? This water comes from a steam intensifier; the area on which the steam acts is 300 square inches; the area of the ram is 30 square inches. What is the pressure difference of the steam, neglecting friction? (I.C.E., *Oct.*, 1900.)

17. Describe carefully the action of the air vessel on the delivery side of a vertical single-action bucket-pump with three sets of valves, suction, bucket, and delivery. Under what circumstances would you consider an air vessel on the suction side useful? Under what circumstances would you expect that the suppression of the suction-valves would improve the delivery? (I.C.E., *Oct.*, 1900.)

18. Give an expression for the tension of the bucket-rod or piston-rod in a common suction pump. The barrel of the pump is 8 inches diameter and the stroke of the bucket or piston is 10 inches. Find the work done per stroke in foot-pounds when the spout is 16 feet above the surface of the water in the well. (I.C.E., *Oct.*, 1900.)

19. Explain the limits of action of the common lifting pump and find the tension of the bucket rod at any point of the upward stroke when the pump is in full working. (I.C.E., *Feb.*, 1901.)

20. A vertical sluice gate of rectangular form closes a channel which is 6 feet in width. On one side of the gate the water stands 5 feet above the cill, and on the other side only 2 feet above the cill. Find the horizontal pressure which is exerted upon the gate as a whole and resisted by its supports. (I.C.E., *Feb.*, 1901.) *Ans.* 4,687·5 lbs.; 750 lbs.; total pressure on gate = 3,937·5 lbs.

21. In the case described in the previous question, find the centre of action of the whole horizontal pressure. (I.C.E., *Feb.*, 1901.) *Ans.* x = 1·86 feet from the bottom.

22. The inner face of a masonry dam is a plane vertical surface 12 feet in height. The section of the dam is a right-angled triangle 9 feet wide at the base, and the weight of the masonry is 125 lbs. per cubic foot. When the reservoir is filled to the upper edge of the dam, find the point at which the resultant line of pressure intersects the base. (I.C.E., *Feb.*, 1901.) *Ans.* 5 feet $7\frac{1}{2}$ inches from the vertical face of the dam.

23. In designing a hydraulic forging press to exert a maximum pressure of 3,000 tons what water pressure would you recommend; how many hydraulic cylinders would you use; would you cast these in steel or iron; and what thickness would you give them? (I.C.E., *Feb.*, 1901.)

24. Write a short essay on the special advantages of hydraulic over other kinds of cranes for dock work. (I.C.E., *Feb.*, 1901.) (*Refer back to Lecture X., Vol. I., before answering this question.*)

25. A box in the form of a cube, of internal dimensions 1 foot, has its base horizontal and is half filled with water. One vertical side is kept in its position by four screws only, one at each angular point. Find the tension in these screws due to the water pressure. (I.C.E., *Feb.*, 1901.)

26. State what is meant by the centre of pressure of a plane immersed in water. A rectangular door 10 feet long and 4 feet broad is placed in the side of a tank with its plane vertical, and one of its long edges at the surface of the water. It is hinged at this edge. Find the horizontal force at the lower edge to keep the door in its vertical position (1 cubic foot of water weighs $62\frac{1}{2}$ lbs.). (I.C.E., *Oct.*, 1901.) *Ans.* $3{,}333\frac{1}{3}$ lbs.

27. Show how the gland in a plunger-pump may be packed. Sketch suction and delivery valves suitable for such a pump delivering 50,000 gallons per hour against a head of 300 feet. (I.C.E., *Oct.*, 1901.)

28. The ram of a hydraulic accumulator is 10 inches diameter, and has a stroke of 11 feet. When fully loaded the water-pressure is 800 lbs. per square inch. If the whole energy of the accumulator water could be used in three minutes, what horse-power would it exert? (I.C.E., *Feb.*, 1902.)

29. If the area of a horizontal section of a ship at the water-line is 15,000 square feet, and the sides vertical where they cut the water, find the extra depth the ship will sink when loaded in fresh water with 750 tons of cargo. What depth would the ship sink if floating in salt water of specific gravity 1·026 when loading? (1 cubic foot of fresh water weighs 62·5 lbs.) (I.C.E., *Feb.*, 1902.) *Ans.* $x = 1$ foot $9\frac{1}{2}$ inches in fresh water, and 1 foot 9 inches in salt water.

30. A flap valve is 2 feet square, and stands inclined at 60° to the horizontal. Find the force needed to open this valve when the sea-water level is 6 feet above the horizontal hinge of the valve. The force to open the valve is applied at right angles to an arm 16 inches long attached at right angles to the upper edge of the valve close to the hinges. (I.C.E., *Feb.*, 1902.)

31. Explain the construction of the common type of hydraulic jack. (I.C.E., *Feb.*, 1902.)

32. Give a rough diagram showing the construction of the ordinary form of Worthington pump. (I.C.E., *Feb.*, 1902.)

33. A rectangular area is immersed in water with one edge in the surface; express the resultant fluid pressure upon one side of it, and state where this resultant acts. The width of a lock is 12 feet, and that of each gate $6\frac{1}{4}$ feet; if the height of the water be 10 feet inside the lock and 4 feet outside, find the resultant fluid pressure on each gate, and also the pressure (assumed to be acting symmetrically) between the two gates. (I.C.E., *Oct.*, 1902.)

34. Give sketches and describe the construction of a hydraulic crane. Estimate the volume of the ram necessary, if a weight of 5 tons is to be lifted 20 feet, the water-pressure being 700 lbs. per square inch and efficiency of machine $\frac{1}{2}$. (I.C.E., *Oct.*, 1902.) *Ans.* $1\frac{1}{9}$ cubic feet.

N.B.—*See Appendices B and C for other questions and answers.*

LECTURE II.

CONTENTS.—Frictional Resistances and Efficiencies of Machines in General—Example I.—Application to the Steam Engine—Efficiency of a Reversible Machine—Example II.—Movable Hydraulic Cranes—Movable Electric Crane—Details of Hoisting Brake and Levers for Working the 3-Ton Electric Crane—Abstract of Report, Tables and Curves on Comparative Trials for Efficiency of 3-Ton Hydraulic and Electric Cranes—Relative Cost of Hydraulic and Electric Power—Crane Tests—Explanation of Efficiency Curves—Electric Cranes for Manchester Ship Canal—Questions.

Frictional Resistances and Efficiencies of Machines in General.—In Lecture VII., Vol. I., we showed how to calculate the work lost in friction in the cases of plane and cylindrical surfaces. As these are the principal kinds of rubbing surfaces in machinery, we might, if we knew the exact pressures between the various surfaces, calculate the total frictional losses and thus find the efficiency of the machine. But there are other losses of work during its transmission, such as the bending of ropes, belts, chains, &c., which it is almost impossible to calculate with exactness. Even if we could calculate all these various losses, the task would be a most tedious and unprofitable one. Hence, in finding the efficiency of any machine, we have recourse to direct experiment on the machine as a whole, the results of which furnish us with data from which we can determine the exact efficiency. In general, however, a machine will not have the same efficiency when working under different loads, owing to the fact that the frictional and other resistances are not proportional to the efforts and loads. To make this clearer, suppose we take the case of a steam engine. Here the friction between the piston and its cylinder, the piston-rod, valve rods, &c., and their stuffing boxes, as well as the friction at the journals due to the weights of the flywheel and shaft, are, severally, constant in amount, whether the engine be developing full power or running light. Hence, in all machines there is a certain proportion of the frictional and other resistances constant, whatever be the magnitude of the effort and the load. This (as just explained in the case of the steam engine) is due to forces between the parts of the machine itself, such as the weights and inertia of the moving parts, or the resistances offered to the bending and stretching of parts, and which have little or no connection with the effort exerted or the load overcome. From these facts we see that the lost work will be proportionally less, and the useful work proportionally more the greater the total work expended. In other words, *the efficiency of a machine will, in general, be higher the greater the load.* This statement, however, is not true for hydraulic and other machines where fluid resistances occur, or where the speed of the machinery is *very* great. As will be seen later on, fluid resistances increase very rapidly with the speed. In high-speed machinery the effects of the inertia of the moving parts introduce other serious losses which must not be ignored when calculating their efficiency.

The "Principle of Work," when applied to a machine, has already been written in the form :—

Total work expended = Useful work done + Lost work.

Or, $W_T = W_U + W_L$ [see eqn. (I.), Lect. IV.]

The last term, W_L, is made up of two distinct parts—one part depending on W_T and W_U, and a second part which is constant, and, therefore, independent of W_T and W_U. Hence, we may put :—

$$W_L = \mu_1 W_T + \mu_2 W_U + C.$$

Where μ_1 μ_2 are numerical coefficients, and $\mu_1 W_T$, $\mu_2 W_U$ are the frictional resistance due to the effort and load applied; while C represents an amount of lost work which is constant for the same machine

$$\therefore \quad W_T = W_U + \mu_1 W_T + \mu_2 W_U + C.$$

$$\text{Or,} \quad (1 - \mu_1) W_T = (1 + \mu_2) W_U + C. \quad . \quad . \quad . \quad . \quad . \quad (I)$$

This is a *general* equation for the "Principle of Work" as applied to machines. In most machines the coefficient, μ_1, which depends on the effort, Q, must necessarily be very small, and may sometimes be neglected.

Dividing both sides of the equation (I) by $(1 - \mu_1)$ we get :—

$$W_T = \frac{1 + \mu_2}{1 - \mu_1} W_U + \frac{C}{1 - \mu_1},$$

$$,, \quad = \left(1 + \frac{\mu_1 + \mu_2}{1 - \mu_1}\right) W_U + F.$$

$$\text{Or,} \quad W_T = (1 + f) W_U + F. \quad . \quad . \quad . \quad . \quad . \quad (II)$$

Where $f = \left(\frac{\mu_1 + \mu_2}{1 - \mu_1}\right)$ and $F = \left(\frac{C}{1 - \mu_1}\right)$ are new constants derived from the old ones, as shown.

For some purposes, it is more convenient to write equation (II) in the following forms :—

$$W_T = k W_U + F. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (III)$$

$$\text{Or,} \quad Q x = k W y + F. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (IV)$$

Where (as in Lecture IV., Vol. I.) x and y are respectively the displacements of the effort Q and the load W, in a given time.

Suppose the machine to run light. Then $W = 0$, and Q_0 is the effort required to drive the machine.

From equation (IV), we get :—

$$Q_0 x = F.$$

That is, F *represents the work done in driving the machine unloaded.*

Dividing both sides of this equation by x, we get :—

$$Q_0 = \frac{F}{x}.$$

This is the effort required to drive the machine light.

Substituting the above value for F, in equation (IV), we get :—

$$Q x = k\ W y + Q_0 x.$$

Or,

$$Q = k\ W \frac{y}{x} + Q_0$$

$$\therefore \qquad Q = k\ W \frac{v}{V} + Q_0. \qquad \text{(V)}$$

This is a *general* equation connecting the effort Q and the load W for any machine. Since the velocity ratio, $\frac{V}{v}$, is constant for the same machine, we might write the last equation in this convenient form :—

$$Q = K\ W + Q_0. \qquad \text{(VI)}$$

Where, $K = k \Big/ \frac{V}{v}$, and can be found by experiment for any machine.

Example I.—In an ordinary block and tackle having three sheaves in the upper and two in the lower block, it is found by experiment that a force of 11 lbs. is required to lift a weight of 40 lbs., and a force of $24\frac{1}{2}$ lbs. to lift a weight of 100 lbs. Find a general expression for the relation between Q and W in this arrangement, and the weight which could be raised by a force of 56 lbs. Find, also, the efficiency of the machine in all three cases, and the actual mechanical advantage.

Answer.—The general relation between Q and W must be of the form :—

$$Q = K\ W + Q_0. \qquad (1)$$

From the results of the first experiment, we get :—

$$11 = 40\,K + Q_0. \quad \ldots\ldots \quad (2)$$

And from the results of the second experiment, we get :—

$$24\tfrac{1}{2} = 100\,K + Q_0. \quad \ldots\ldots \quad (3)$$

(3)–(2) $$13\tfrac{1}{2} = 60\,K,$$

$$\therefore \quad K = \frac{27}{120} = \frac{9}{40}.$$

Substituting this value of K in equation (2), we get :—

$$11 = \frac{9}{40} \times 40 + Q_0,$$

$$\therefore \quad Q_0 = 2 \text{ lbs.}$$

Or, the effort required to drive the machine light is 2 lbs.

Substituting the values of K and Q_0 in equation (1), we get :—

$$Q = \frac{9}{40}\,W + 2 \quad \ldots\ldots \quad (4)$$

which is the general formula required.

To find the weight which could be lifted by an effort of 56 lbs., we substitute this value in equation (4), and get :—

$$56 = \frac{9}{40}\,W + 2,$$

$$\therefore \quad W = \frac{54 \times 40}{9} = 240 \text{ lbs.}$$

The efficiency of the machine in any case is found from the usual formula, viz. :—

$$\textit{Efficiency} = \frac{\text{Useful work done}}{\text{Total work expended}} = \frac{W\,y}{Q\,x} = \frac{W}{Q}\,\frac{v}{V}.$$

Since there are five ropes supporting the weight, W, in this system of block and tackle, it is clear that :—

$$\frac{v}{V} = \frac{1}{5}.$$

$$\therefore \quad \text{Efficiency} = \frac{1}{5}\,\frac{W}{Q}.$$

Then :—

$$(1 - \mu_2)\, W\, y = (1 + \mu_1)\, w\, x + C. \quad . \quad . \quad . \quad . \quad (2)$$

Now subtracting equation (2) from (1), in order to eliminate C, we get :—

$$(1 - \mu_1)\, Q\, x - (1 - \mu_2)\, W\, y = (1 + \mu_2)\, W\, y - (1 + \mu_1)\, w\, x$$

$$\therefore \qquad (1 + \mu_1)\, w\, x = 2\, W\, y - (1 - \mu_1)\, Q\, x.$$

Dividing both sides by $(1 + \mu_1)$ and $W\, y$, we get :—

$$\frac{w\, x}{W\, y} = \frac{2}{1 + \mu_1} - \left(\frac{1 - \mu_1}{1 + \mu_1}\right) \frac{Q\, x}{W\, y}.$$

$$\textit{Efficiency when reversed} = \frac{\text{Useful work done in raising } w}{\text{Total work expended by } W}.$$

$$\text{,,} \qquad = \frac{w\, x}{W\, y},$$

$$\text{,,} \qquad = \frac{2}{1 + \mu_1} - \left(\frac{1 - \mu_1}{1 + \mu_1}\right) \frac{Q\, x}{W\, y}.$$

But $\dfrac{W\, y}{Q\, x}$ = the *original efficiency* of the machine, or efficiency when working in the usual manner.

$$\therefore \quad \left.\begin{array}{r}\textbf{Efficiency of Machine}\\ \textbf{when Reversed}\end{array}\right\} = \frac{2}{1 + \mu_1} - \left(\frac{1 - \mu_1}{1 + \mu_1}\right) \times \frac{1}{\eta} \quad . \quad \text{(VII)}$$

where η denotes the original efficiency of the machine.

It is clear that the machine will not reverse unless the above efficiency be greater than 0—

$$\textit{i.e.}, \text{ unless} \quad . \quad . \quad \frac{2}{1 + \mu_1} - \left(\frac{1 - \mu_1}{1 + \mu_1}\right) \times \frac{1}{\eta} > 0.$$

$$\textit{i.e.}, \text{ unless} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \frac{2}{1 + \mu_1} > \left(\frac{1 - \mu_1}{1 + \mu_1}\right) \times \frac{1}{\eta}.$$

$$\textit{i.e.}, \text{ unless} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \eta > \tfrac{1}{2}(1 - \mu_1).$$

Consequently, the machine will not reverse until the original efficiency, η, be greater than $\frac{1}{2}(1 - \mu_1)$, and, if it be less than this, it will not reverse even if the original hauling force, Q, be entirely withdrawn. For, if $\eta = \frac{1}{2}(1 - \mu_1)$, the efficiency of the machine when reversed would vanish (as may be seen by substituting this value in equation (VII)). If $\eta < \frac{1}{2}(1 - \mu_1)$, the efficiency would be negative, which is absurd.

We have already stated that in most machines the fraction, μ_1, is *very small*, and may be neglected. Hence we get the important statement that

A machine will not reverse even when the hauling force is entirely withdrawn, if the efficiency is less than $\frac{1}{2}$ or 50 per cent.

Example II.—Apply the "Principle of Work" to calculate the relation between P and W in the screw lifting jack, fitted with screw and worm-wheel gear. The handle which works the jack has a radius of 14 inches, pitch of screw 1 inch, number of teeth on worm-wheel 20, and the worm is double threaded; find the force which must be applied at the end of the handle in order to raise a weight of 4 tons, friction being neglected. If the actual force required to raise this weight be 40 lbs., what is the efficiency of the apparatus?

Answer. — Let L = length of handle, p = pitch of screw, N = number of teeth on worm-wheel, n = number of threads on worm.

(1) Suppose the handle to make one complete turn. Then, since there are n threads on the worm, and N teeth on the worm-wheel, it is clear, that for one turn of the handle, the worm-wheel, which forms the nut of the screw, will have made $\frac{n}{N}$ part of a complete turn. Hence the weight, W, will have been raised through a height $\frac{n}{N} \times p$.

∴ By the Principle of Work, we get :—

P × its displacement = W × its displacement.

$$\therefore \qquad P \times 2\pi L = W \times \frac{n}{N} \times p,$$

$$\therefore \qquad \frac{P}{W} = \frac{np}{2\pi L N}.$$

(2) In the example $p = 1''$, $L = 14''$, $n = 2$, $N = 20$, $W = 4 \times 2{,}240$ lbs.

$$\therefore \qquad P = 4 \times 2{,}240 \times \frac{2 \times 1}{2 \times \frac{22}{7} \times 14 \times 20} \text{ lbs.},$$

$$\text{,,} = 10{\cdot}18 \text{ lbs.}$$

(3) $$\textit{Efficiency} = \frac{P}{Q} = \frac{10{\cdot}18}{40} = {\cdot}2545, \text{ or } 25{\cdot}45 \text{ per cent.}$$

Movable Hydraulic Cranes.—The following extracts from the "General Specification" by Mr. George H. Baxter, Chief Mechanical Engineer to the Clyde Navigation Trustees, are herewith reproduced along with the accompanying reduced drawings, in order to enable the student to form an idea of the main requirements fulfilled by these double-power 5 and 3-ton hydraulic cranes at the Prince's Dock, Glasgow. The cranes were made by A. Chaplin & Co., Mechanical Engineers, Govan, and have been constantly in use since 1898. Interesting comparative tests were made in March, 1903, between one of these cranes working with its 3-ton lifting ram, and a 3-ton electrical crane, which are described immediately after their specifications. These tests afford useful information regarding the two systems of power supply :—

General Description of Cranes.—"Each crane to be mounted on a steel travelling carriage, with arch for passage of locomotives, having a rail gauge of 14 feet between centres of double rails, which are to be provided and laid on quay and cope of quay wall by the Trustees. Jib to be capable of revolving through at least one and a-quarter turns. The cranes to be double-powered, to lift maximum loads of 5 tons and 3 tons. All the lifts to be by single chain, and the working pressure to be 750 lbs. per square inch. The cranes to have a factor of safety throughout of not less than 8 to 1, excepting the hoisting chains."

Carriages.—"The carriages to be supported by four wheels of cast iron, bushed with gun-metal $\frac{5}{8}$-inch thick, with wrought-steel weldless tyres shrunk on each side of central flange after being turned and bored. The axles to be turned steel forgings, secured by strong split cotters. Brackets carrying wheels to be so arranged that the wheels can be easily removed. Shackles and steadying screws to be provided at each corner. The outer side of carriage to be kept at least one foot back from edge of cope, and a hand-rail to be run (the whole length of carriage) outside for convenience in shifting mooring ropes."

Range of Lift.—"Each of the cranes to have a total range of lift of 75 feet, being 35 feet above the cope to 40 feet below, and the chain to project 32 feet beyond face of quay wall when the jib is square to it."

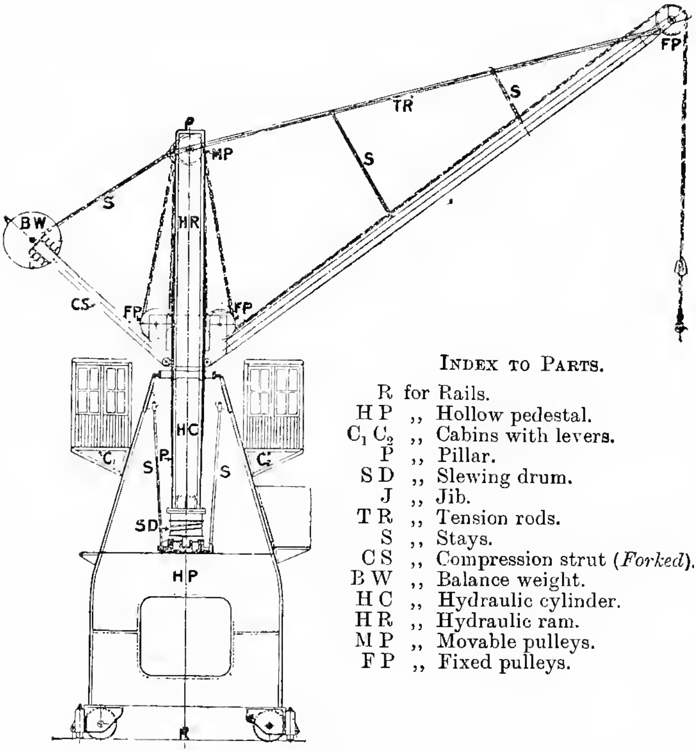
Jib and Counter-balance Weight.—"The jib of each crane to have a rake of not less than 41 feet, and the jib-head pulley to be 60 feet from level of cope of quay wall. Jib stays to be secured independently of pins on which jib-head pulley revolves, so that the latter may be taken down without disturbing the stays. A steel bracket on hinged joint, or other means, to be provided for locking jib when cranes are not in use. Wrought-iron ladder to be fitted to each jib to give access for lubricating jib-head pulley. Counter-balance weight to be high enough to clear roof of shed."

Houses for Driver.—"A house to be provided for the driver on each side of upper part of carriage. Handles for working to be conveniently placed, and marked for hoisting, lowering, and slewing."

Platforms, Hand-rails, Ladders, Guards, &c.—"Platforms, hand-rails, gangways, ladders, footsteps, &c., to be fitted. Suitable lock-up doors and other arrangements to be made to give access to all chains, pulleys, bearings, and other parts, for lubrication and repair. Guards for chains and rollers or rubbing pieces to be fitted wherever required. Elm sheathing, $1\frac{1}{2}$ inches thick, to be laid on inside of jib, and secured thereto by bolts and nuts."

Chains.—"The hoisting chains to be 1 inch diameter and tested in accordance with the standard requirements of the Clyde Navigation Trustees. Each hoisting chain to have a swivel hook of special form and strength, shown by the drawing. A balance ball to weigh at least $2\frac{1}{2}$ cwts., bolted on in halves a few feet above the hook. Also, a wrought-iron or steel clip of ample strength, bolted to standing part of chain, for adjusting height of hook."

Pulleys.—"The standing pulleys to be of the diameter shown in the drawings, and to be arranged on either side of ram as in hydraulic cranes at Queen's Dock. Suitable guards for chain to be provided. Pins for pulleys and carriage wheels to be bored longitudinally, with oil cup in one end and oil hole to each pulley and wheel. All pulleys to be bushed with gun-metal, and be fitted with two steady pins tapped through boss."



INDEX TO PARTS.

R	for	Rails.
H P	,,	Hollow pedestal.
C_1 C_2	,,	Cabins with levers.
P	,,	Pillar.
S D	,,	Slewing drum.
J	,,	Jib.
T R	,,	Tension rods.
S	,,	Stays.
C S	,,	Compression strut (*Forked*).
B W	,,	Balance weight.
H C	,,	Hydraulic cylinder.
H R	,,	Hydraulic ram.
M P	,,	Movable pulleys.
F P	,,	Fixed pulleys.

GENERAL VIEW OF MOVABLE HYDRAULIC CRANE.
(Made by A. Chaplin & Co., Govan, for the Clyde Trust.)

Valves.—"The working valves for hoisting, lowering, and slewing to be

Pipes and Cocks.—"All pipes exposed to the full working pressure to be of strong lap-welded hydraulic tubing of the required thickness, with flanges of cast-iron, turned and faced, with recessed joints to standard sizes. Drain cocks of gun-metal to be fitted to all hoisting and slewing cylinders and all pipes at lowest parts, so as to thoroughly drain them. Air cocks to be fitted to all cylinders and pipes as high as practicable."

Travelling Gear.—"Efficient hand travelling gear to be provided on each crane, so that it may be quickly and easily transported along the quay by not more than six men."

Gas Fittings.—"To prevent damage by frost, each crane to be provided with two gas stoves and burners, including wrought-iron pipes, cocks, and other fittings, for hoisting and slewing cylinders, pipes, &c. A suitable gas stove, with all necessary connections, to be supplied in each crane house. Wrought-iron gas piping, with cock and connection for india-rubber tubing, to be fitted on each carriage, for supplying stoves, &c."

Movable Electric Crane.—The following extracts from the "General Specification," by George H. Baxter, Chief Mechanical Engineer to the Clyde Navigation Trustees, are herewith reproduced, along with the accompanying reduced drawings, in order to enable the student to form an idea of the main requirements fulfilled by the 3-ton movable electric crane, as built and erected at Prince's Dock, Glasgow, by Stothert & Pitt, Bath.* After the specification there follows the data of the comparative tests between this crane and the previously described hydraulic one:—

General Description.—"The crane is to be movable, mounted on a high portal carriage to run on a line of double rails, 14 feet apart between centres, laid along the front of the South Quay, Prince's Dock, Glasgow Harbour. The crane is intended to be used for carrying out an extensive series of tests for the purpose of comparing the relative efficiencies of electric and hydraulic cranes of similar capacity under varying conditions and over a prolonged period. Provision is, therefore, to be made for applying and fitting up suitable recording instruments for the measurement of electric energy delivered to the hoisting and slewing motors, and of the mechanical energy developed by the crane."

General Dimensions.—"The following are the general dimensions of the crane:—

Maximum working load, ...	3 tons.
Total range of lift, ...	80 feet.
Radius of lifting rope, ...	41 ,,
Projection of rope beyond face of quay, . . .	32 ,,
Centre of jib-head pulley above cope, . . .	60 ,,
Clear lift above cope of quay, ...	50 ,,
Extreme radius of any revolving part at back of crane, .	9 ,,
Clear height from cope to lowest revolving part of crane,	29 ,,

Speeds.—"The lifting speed with a load of 3 tons to be 150 feet per minute, the speed with smaller loads to be correspondingly greater. The slewing speed measured at the hook to be 300 feet per minute with maximum load."

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* The electric motor, volt and ampere metres for this crane were made by Siemens Brothers & Co., Ltd., of Woolwich and London.

Factors of Safety and Stability.—"The carriage, crane framing and jib, gearing and wire rope, to have a factor of safety of 10 to 1. Stability to be maintained with a load of 6 tons suspended in the crane hook."

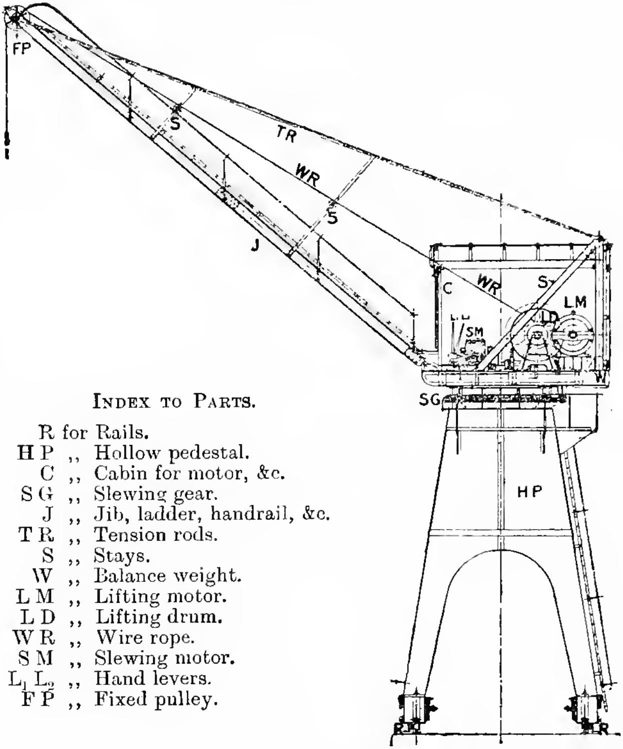
Carriage.—"The carriage is to be constructed with an arch 15 feet high, sufficiently wide for the passage of locomotives. To be made of Siemens steel plates, angles, and bars. The wheel base to be 14 feet. The carriage to be supported by four strong cast-steel wheels turned to 2 feet 6 inches diameter on tread, and having a central flange, 1 inch deep by 1¾ inch wide. The wheel bosses to be bored, faced, and bushed with gun-metal, ⅝ inch thick. The axles to be turned steel forgings, secured by strong split cotters; oil holes to be drilled through centre with short bent lubricating pipes screwed into inner ends. The length and diameter of axle bearings to be specified or shown on plan. Brackets carrying wheels to be of approved design, and so arranged that the wheels can be easily removed if required. Shackles and steadying screws similar to those on hydraulic cranes to be provided at each corner. The side of carriage next dock to be kept at least 1 foot back from the edge of the quay, and a hand-rail to be run the whole length of carriage outside for convenience in shifting mooring ropes. Provision to be made on side of carriage for connecting the motors with the main circuit conductors."

Framing Jib, &c.—"The framing of the revolving part of the crane, jib, and back stays to be made of steel plates and angles, channels, or tee bars carefully fitted and riveted together. The jib stays to be steel forgings or bars drawn or shaped out of the solid without welds, and to be secured by turned pins independently of the pin on which the jib-head pulley revolves, so that the latter may be removed without disturbing the stays. An 18-inch wrought-iron ladder, with hand-rail on each side, to be fitted to jib to give access for lubricating the jib-head pulley. The pulley to be of cast steel, turned in groove to not less than 24 times the diameter of hoisting rope. To be provided with rope, guides, and guards. The pin to run in adjustable gun-metal bushes, and the bearings to have fixed lubricators of approved pattern. Counterbalance weights to be provided in back end of framing to balance the jib and the load."

Centre Post, Roller Path, and Rollers.—"A strong centre post of forged ingot steel, with hole drilled throughout its length for the conductors to pass through, to be turned all over and fitted to strong cast-iron bed plate, or sockets secured to upper part of carriage by turned and fitted bolts. If a roller path is fitted, it is to be a solid ring of Siemens steel accurately turned and faced flat, and secured to carriage by turned and fitted bolts. The rollers are to be of cast steel, truly turned, bored, and keyed to fit strong steel axles, having large bearing surfaces working in gun-metal bushes, adjustable by bolts and nuts, and to be easily removable for repairs."

Plating and Riveting.—"All plate edges are to be planed, all butts to be accurately fitted, and ends of bars neatly trimmed. Rivet holes in framing, jib, &c., are to be drilled in place, and all holes to be made fair without drifting. All rivets to be snap-headed outside."

Hoisting Winch.—"To be single-geared, driven by a separate motor. The bed-plate to be of cast iron, planed to receive the side cheeks and hoisting motor. One of the cheeks to be of box form, planed, bolted together in halves, jointed, and made oil tight; the lower half to form an oil-bath for main spur wheel and the pinion on armature shaft. The barrel to be of cast iron, 24 times the diameter of the hoisting rope, and to be turned with a continuous spiral groove, long enough to take the whole



INDEX TO PARTS.

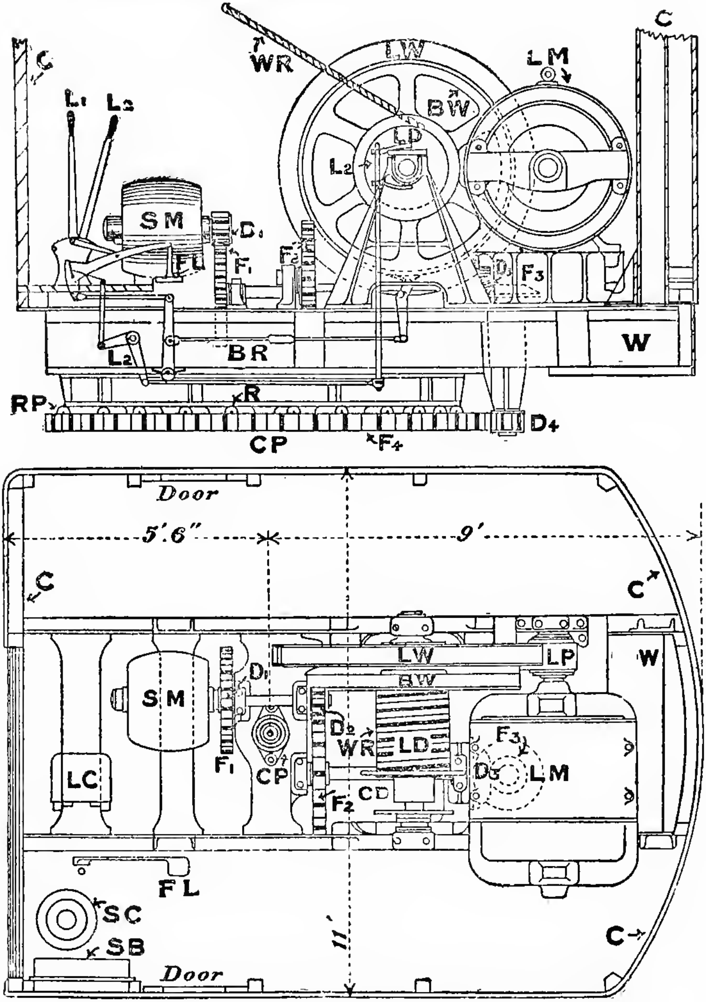
R for Rails.
H P ,, Hollow pedestal.
C ,, Cabin for motor, &c.
S G ,, Slewing gear.
J ,, Jib, ladder, handrail, &c.
T R ,, Tension rods.
S ,, Stays.
W ,, Balance weight.
L M ,, Lifting motor.
L D ,, Lifting drum.
W R ,, Wire rope.
S M ,, Slewing motor.
L_1 L_2 ,, Hand levers.
F P ,, Fixed pulley.

GENERAL VIEW OF A 3-TON MOVABLE ELECTRIC CRANE.
(Made by Stothert & Pitt, Bath, for the Clyde Trust.)

INDEX TO PARTS OF LIFTING, LOWERING, AND SLEWING GEAR.

(*See Section and Plan on opposite page.*)

C for Cabin.
S B ,, Switch board.
L_2 ,, Lever for clutch, C D.
C D ,, Coil-clutch and disc.
L C ,, Lifting controller.
L M ,, ,, motor.
L P ,, ,, ,, pinion.
L W ,, ,, ,, wheel.
L D ,, ,, ,, drum.
W R ,, Wire rope.
F L ,, Foot-brake lever.
B R ,, Brake rod.
B W for Brake wheel.
L_1 ,, Lever for slewing brake.
S C ,, Slewing controller.
S M ,, Slewing motor.
D_1 D_2 D_3 D_4 ,, Driving pinions.
F_1 F_2 F_3 F_4 ,, Follower wheels.
R P ,, Roller path.
R ,, Rollers.
C P ,, Central pivot.
W ,, Balance weight.



Vertical Section and Plan of Cabin and Driving Gear
For the 3-Ton Electric Crane belonging to the Clyde Trust.

single rope. The main spur wheel is to be of cast steel, the pinion of forged steel, both to have wide teeth of suitable pitch, accurate form, and machine cut. The ratio of gearing and diameter of barrel and jib-head

Hoisting Brake.—"An automatic brake to be provided. This brake to be worked also by a hand or foot lever, and to effectually hold the maximum load at any point. It must be so designed, that when the current is switched on to the hoisting motor, the brake is automatically released the moment hoisting begins. And, in switching off, or in the event of failure of current, the brake would be put in gear automatically, and without shock." (*See the pages describing construction and action of this brake at the end of this Specification.*)

Hoisting Rope.—"To be made of the highest quality of plough steel wire, specially flexible and free from twisting. The rope to be provided with an overhauling weight in halves, and an eye with solid wrought-iron thimble at end, bored to fit the shackle-pin; also a 3-feet length of $\frac{13}{16}$-inch short-link crane chain, and swivel hook of approved design. The ultimate tensile strength of the wire (which is to be tested and approved of before the rope is made) is not to exceed 110 tons per square inch."

Slewing Gear.—"To be worked by separate motor, having a wrought-steel worm keyed on end of armature shaft, gearing into a phosphor-bronze worm wheel on vertical shaft, with pinion at lower end in gear with the slewing wheel, which is to be secured to top of carriage with turned and fitted bolts. The worm to have large thrust bearings of gun-metal, and to run in an oil-bath with dustproof cover. The vertical shaft to be of large diameter to ensure stiffness, and to have collars forged on at ends. The bearings are to be of gun-metal, adjustable by means of bolts and nuts. All teeth to be of suitable pitch, accurate form, and machine cut. A brake worked by foot lever to be provided to control the speed at any point."

Travelling Gear.—"The crane to be provided with efficient travelling gear fitted to each carriage wheel, worked by hand, and so connected by means of clutches that it may be disengaged to admit of the crane being moved along the quay by a hydraulic capstan. Spur wheels on axles of carriage wheels to be 6 inches clear of the ground."

Electric Motors.—"Both motors are to be of the drum armature enclosed type. They are to be series-wound, and constructed to work on a continuous current circuit at an E.M.F. of 230 volts, but capable of working at 260 volts if required. The hoisting motor to be 50-brake H.P., and slewing motor 10-brake H.P. Both are to be designed to give every facility for examination and repairs. The armatures to be of the strongest form of construction, well insulated throughout, with provision for thorough ventilation. The rise of temperature in the armature must not exceed 60° F. above the surrounding atmosphere. The commutators are to be made of pure annealed copper bars, insulated with mica, and to have large wearing surfaces, with self-adjusting carbon brushes. The armature shafts to be of forged steel, working in long bearings lined with white metal, bored out perfectly smooth and true, and provided with efficient automatic lubricating apparatus of the ring type. Chambers to be formed at the ends of each bearing to hold waste oil, with gauge-cocks, drain-cocks, and pipes led to an oil reservoir."

Controlling Switches.—"These are to be of the most improved type, with graduated resistances suitable for cranes. One controlling switch is to be provided for each motor, and of such dimensions as shall prevent undue heating when the crane is working continuously for six hours. The switch contacts must be large and carefully constructed, so as to prevent sparking. These are to be enclosed in an iron casing, with covers and doors for examination and repairs. The controlling switches to be so constructed that the motors may be started, regulated, stopped, reversed, or ran at varying speeds, with varying loads in the simplest and most reliable manner. The regulating handles are to be conveniently arranged in such a position that the driver may see the load during the whole time it is suspended in the crane, and the handles should be so connected up, that they move in the direction it is desired to move the load."

Switchboard, Switches, and Cut-outs, &c.—"A double-pole switch of the quick break lever type and two single-pole cut-outs for the hoisting and slewing motors, together with separate switches and cut-outs for the electric lighting and heating circuits, to be all arranged on a suitable switchboard of slate or marble, framed in channel iron $\frac{3}{16}$ of an inch thick lined with teak. The switches and cut-outs to be properly enclosed and protected, so as to prevent risk of fire or shock in any way, and to be conveniently placed within easy reach of the driver. Sweating sockets to be provided, to which the cables and wiring are to be connected. The cut-out terminals shall be so arranged that the fuses of flat tinfoil may be easily and securely fitted by the craneman. The securing nuts are to be of bright mild steel, and the fuse break not less than 5 inches. Each switch and instrument shall be capable of conveying the maximum current continuously, without sensible rise of temperature. The covered flexible connections and cables shall be of such area, that the current density shall not exceed 800, the gun-metal parts 400, and the effective contact surfaces of switches 200 amperes per square inch."

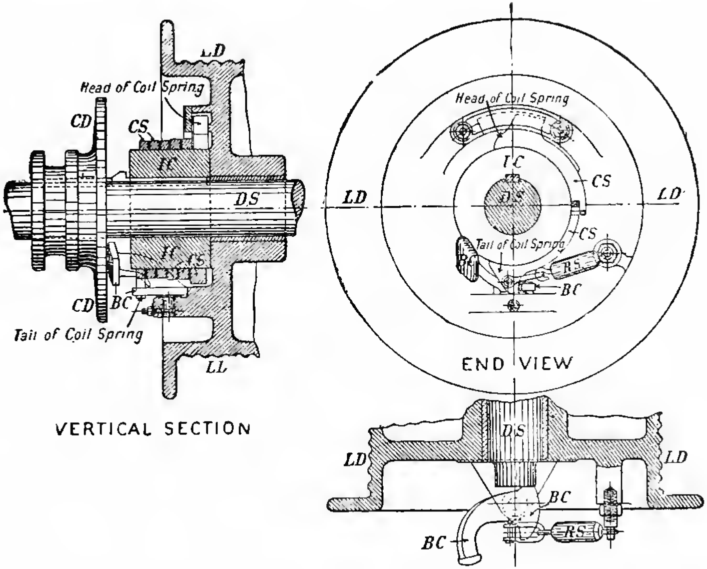
Switchboard Instruments.—"A lamp, an ammeter, a voltmeter, and an automatic circuit breaker (to prevent danger from overloading) to be provided and conveniently arranged on the switchboard, with all necessary connections to the main conductors."

Conductors and Insulation.—"The conductors to be of suitable standard sizes, corresponding to the Cable Makers' Association List. The conductors are to be connected from the terminals of the motors to the controlling switches, from thence to the cut-outs, and therefrom to suitable connections passing through the centre post to sockets at the bottom of crane carriage, and there provided with plug connections to the twin cable. The conductors in the house to be provided with circular sliding contacts and collectors, so as to admit of the crane being slewed in either direction to any extent without difficulty. A portable pair of conductors, with plugs sweated on ends, to be provided to fit sockets on carriage, and also the wall-plugs along the quay. The twin conductor to be of such a length as to extend to a distance of 40 feet on either side of the centre of the crane. The insulation resistance to be not less than 3,000 megohms per mile for the larger, and 4,000 for the smaller conductors."

Wall Plugs.—"Wall-plugs to be fitted in the hydraulic supply hydrant-boxes, or elsewhere, along the quay, with suitable terminals for connection to the electric current supply mains. The boxes to be thoroughly water-tight, and every precaution to be taken against damp."

House.—"A substantially-built timber house to be constructed of sufficient size to give easy access to all parts of the machinery, and ample room for working the crane. A suitable electric radiator, or stove, to be provided, and fitted with a graduated resistance switch, to allow of varying degrees of heat being obtained. Two 32 candle-power Ediswan glow lamps, and two plugs for portable lamps, to be fitted up in house, and a cluster of glow lamps half way up the jib."

Details of Hoisting Brake and Levers for Working the 3-Ton Electric Crane.—In order to make the construction and action



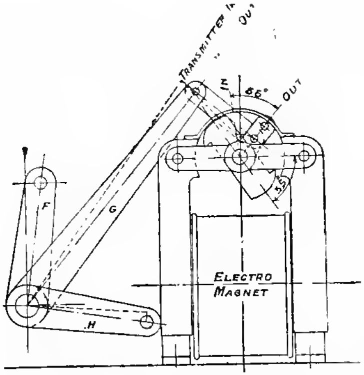
VERTICAL SECTION

END VIEW

SECTIONAL PLAN

DETAILS OF THE COIL CLUTCH FOR 3-TON ELECTRIC CRANE.

L D	for	Lifting drum (*see previous and following views*).
D S	,,	Drum shaft.
I C	,,	Iron chilled cylinder.
C S	for	Coil spring of soft iron.
B C	,,	Bell crank lever.
R S	,,	Releasing spring for B C.
C D	,,	Clutch friction disc.

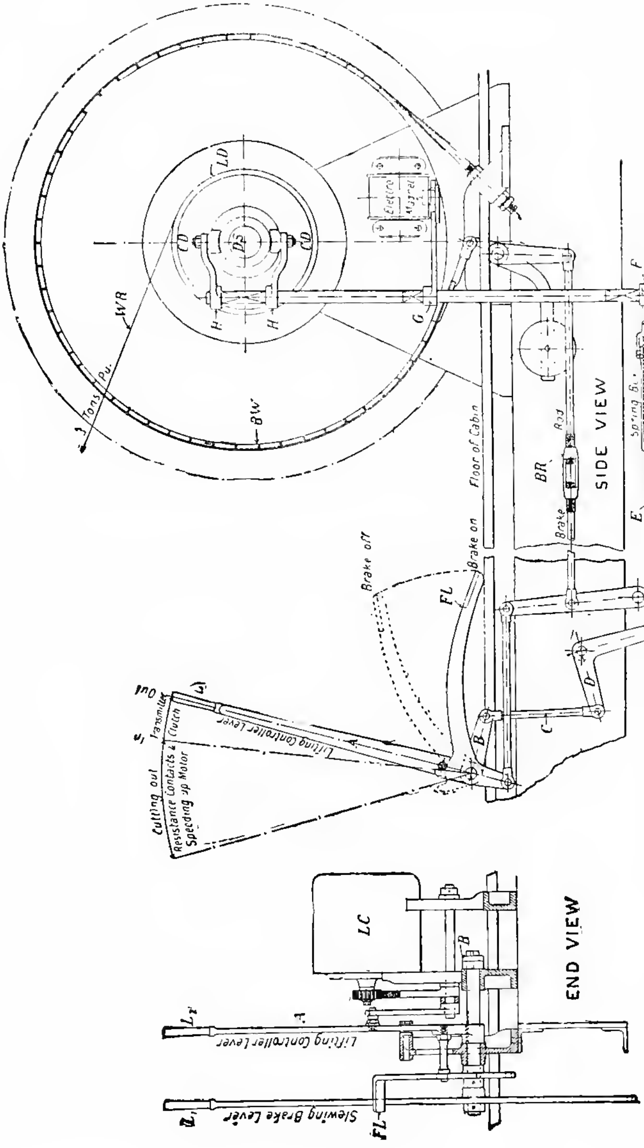


ELECTRO - MAGNET AND LEVERS FOR WORKING COIL CLUTCH.

INDEX TO PARTS OF FIGS

On opposite page and the above one.

L_1	for	Lever to slewing brake.
L_2	,,	Lever to L C and C D.
L C	,,	Lifting motor controller.
A to H	,,	Levers from L_1 to C D.
D S	,,	Drum shaft.
L D	,,	Lifting drum.
W R	,,	Wire rope.
B W	,,	Brake wheel.
B R	,,	Brake rod.
F L	,,	Foot-brake lever to B R.



Levers and Electro-Magnet for Working Stothert & Pitt's 3-Ton Electric Crane.

of these details clear to the student, special views have been herewith reproduced from drawings kindly supplied by the makers.

Coil Clutch.—In the accompanying figures this has been shown by a vertical section, end view, and sectional plan. It will be understood, after comparing the index to parts with these views, that when the clutch friction disc, C D, is pressed forward by the levers, H (shown on the full-page views), it pushes the end of the bell crank lever, B C, so that its inner end pulls upon the tail of the coil spring, C S. This action causes C S to wind close and grip hard upon the iron chilled cylinder, I C. Consequently, we have the lifting drum, L D (which is directly connected to the inner end or head of the coil spring, C S), thereby coupled, through C S, to I C and D S, the drum shaft. The other end of this shaft is keyed to L W, which gears with the lifting motor pinion, L P, and hence with L M, the lifting motor. The load can then be lifted to the desired height.

Braking.—When the clutch friction disc, C D, is disengaged from B C, the latter is brought back to its normal position by the releasing spring, R S. This relieves C S from contact with I C. The lifting drum, L D, with its brake wheel, B W, are thus instantly disconnected from the motor. The load is then, however, held up in position by the brake-strap due to the driver pressing down the treadle or foot lever, F L. The load can now be lowered at any desired speed by simply easing the foot lever, F L, which acts directly on the brake rod, B R, and brake strap of the brake wheel, B W, as shown by the side view on the full-page set of figures. It will thus be seen, that the brake is normally "off," as in a steam crane. The driver lowers a load by pressing his foot more or less on the treadle, thus tightening or slackening the brake-strap in precisely the same way as he would do with a steam crane. The makers admit, that this method of actuating the brake may at first sight appear to be not so automatic or ideal a plan, as that of permitting the weight attached to the brake lever to keep the brake-strap tight on its brake wheel; but, they find it to be much better in actual practice. It will be observed, that if the electric current failed when a load had been raised, and if the driver then stood still and did nothing, the load would undoubtedly drop! However, the current has not been found to fail under these circumstances, and the driver has ample time to press the treadle and check the load.

Lifting.—It will be observed that the first part of the motion of lever L_2 to the left (*i.e.*, from the words "out" to "in"), puts the clutch, C D, into action, and at the same time it brings the transmitter switch of the lifting controller, L C, into circuit with the lifting motor, L M. The remainder of the movement of lever L_2 in the same direction (*i.e.*, to the left) speeds up the motor, L M, by cutting out resistances in the controller of the motor circuit.

Lowering.—On the other hand, the action of putting on or down the treadle foot-brake lever, F L, for the purpose of stopping the motor before lowering a load, automatically throws back the clutch, C D, and disconnects the motor electrically from its controller. The lifting drum, L D, with its attached brake wheel, B W, are now free to move quite independently of the lifting motor and the drum shaft. The load can then be lowered at the desired speed by the required pressure on the foot lever.

Action of the Electro-Magnet.—In the Clyde Trust 3-ton electric crane, Messrs. Stothert & Pitt have adopted an alternative patent arrangement of electro-magnet for engaging and disengaging the clutch friction disc, C D. When it is used, the levers B, C, D, E, F, and spring box (as shown by the side view) are removed. A separate plan of this electro-magnet, with its levers, G and H, are shown. It will be understood, that the current, in passing from the controller, L C, to the lifting motor, L M, traverses and energises the coils of this electro-magnet.

Abstract of Report, Table and Curves on Comparative Trials for Efficiency of the previously-described 3-Ton Hydraulic and Electric Cranes.[*]—The electric crane was erected on the South Quay, Prince's Dock, and started to work on the 24th March, 1902. It has worked since then with the most satisfactory results, and there has been no hitch or trouble of any kind with the electrical machinery. From that date up to the 28th February, 1903, this crane discharged and loaded minerals and general cargo amounting to 43,954 tons, as compared with 40,833 tons dealt with by the 3-ton hydraulic crane alongside of it, whilst working under precisely similar conditions. Both cranes were tested on the 19th and 20th March, 1903, and the results given in tabulated form, indicate that the efficiency of the electric is greater than that of the hydraulic crane at full load, both in working ore and general cargo. The efficiency of the electric crane is also very much greater at the lighter loads, while the cost of producing and distributing electric power to the cranes is less than that of hydraulic power. The curves of performance of these cranes under tests are shown on diagrams 1 and 2. By comparing the corresponding curves, C, on these two diagrams—which show the cost of power used in discharging 1,000 tons of general cargo at varying loads—the difference in favour of the electric crane may be appreciated. For example, in working with loads of

1 ton	2 tons	2½ tons	3 tons

the cost of power used by the hydraulic crane in discharging 1,000 tons of general cargo is

48/3	25/5	20/6	17/2

and by the electric crane

15/11	16/2	16/1½	16/6½

It will be observed, that the costs for the hydraulic crane are nearly in inverse proportion to the loads raised, for the reason that in this type of crane, the same foot-pounds of water are used to lift the empty hook as to lift the maximum load. On the other hand, with the electric crane the cost is practically the same at all loads. This is due to the valuable feature, that the electrical power used, is much more nearly proportional to the power required to raise the load, than is the case with hydraulic cranes. By raising the electrical pressure at the central power-house from 260 volts (its present value) to, say, 500 or 600 volts

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[*] I am indebted to Mr. George H. Baxter, M.I.E.S., for the liberty to make this abstract from his report to the Clyde Trustees.

(as contemplated for the new Clydebank Dock), and by working upon a greatly increased scale, it is probable that the present average cost of 2·558 pence per Unit will be further reduced.*

Relative Cost of Hydraulic and Electric Power.—Since the hydraulic and electric are the only two practicable and available systems of power transmission for dock appliances, such as cranes, capstans, and coal hoists; and, since both systems are equally reliable and convenient, the choice of one or the other depends entirely on which is the more economical, not only in regard to the cost of production and distribution, but also as regards the cost of utilisation.

One system may cost considerably less to produce and distribute power, but in utilisation the gain in point of economy may not be in favour of that system

There are, indeed, so many factors to be taken into account that, for purposes of comparison, and also to determine whether it would cost less or more to purchase or produce power, the cost of each system should be ascertained at three distinct stages, viz. :—(1) The cost of purchase or production. (2) The cost of transmission or distribution. (3) The cost of utilisation.

In order to compare the costs of hydraulic with that of electric power, it is necessary to reduce both systems to a common standard, and to determine in each case the cost of *one horse-power hour* delivered at the appliance where the power is to be utilised. This is the most convenient unit to adopt, and is applicable alike to both systems.

In making up the working costs, the capital expenditure and working costs have been taken from the annual accounts, and embrace all the capital expended, up to the end of 1902, on buildings, foundations, culvert, machinery, and plant for each of the two systems.

The outputs of hydraulic power during the periods under notice, have been found by keeping strictly accurate records of every revolution of the pumping engines. The outputs of electric power have been got from carefully kept daily records of the hours of lighting, pressure, and current used.

The conclusions arrived at are, that electric power costs less to produce, distribute, and utilise than hydraulic power.

Crane Tests.—Tests were conducted under favourable conditions in order to ascertain the following :—

1. The rapidity of working of both types.
2. Their efficiencies.
3. The cost of power used in doing the same amount of work,

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* In a large well-regulated power-house the Cost of Electric Power may be as small as 1d. per Board of Trade Unit—*i.e.*, 1d per 1,000 volt-ampere-hours or watt-hours.

viz., to raise 30 feet, slew 100 degrees, and discharge into waggons 1,000 tons of cargo and 1,000 tons of ore, when working at loads varying from 1 to 3 tons for cargo, and at 2½ and 3 tons for ore.

Particulars of Hydraulic Crane and its Gauges.—The pressure in the hydraulic 7-inch main was maintained at 750 lbs. per square inch at the Power-House. The 3-ton hoisting ram was used. It is 11 inches diameter, and displaces ·66 cubic foot of water per foot of travel, or ·11 cubic foot per foot of hoist, the velocity ratio being 1 to 6. The turning ram is 9 inches diameter, travels 15½ inches, and uses ·57 cubic foot in slewing through 100 degrees. The water used was measured by the displacement of the rams, and 5 per cent. was allowed for slip and leakage.

Pressure gauges were fitted on the main pressure pipe at the hydrant connected to the crane, and to the upper parts of the hoisting and slewing cylinders. The pressures as indicated by these gauges were noted at each lift.

Particulars of Electric Crane and its Meters.—The hoisting gear is single reduction, and is worked by a 50 H.P. Siemens' motor at a pressure of about 240 volts. The pressure at the Power-House was maintained at 260 volts, whilst the current naturally varied with the load. The turning motor is 10 H.P.

The electric crane was equipped with Siemens' volt and ampere meters, and also with Chamberlain & Hookham's watt meters, which were overhauled and found to be accurate before the tests were commenced.

Details of Testing Operations.—In testing for rapidity of working, each crane was provided with two tipping buckets, one for the load and one for the empty bucket. For convenience, the tests were carried out to specific instructions in double cycles, by working the cranes as rapidly as possible for two hours, under each separate load of 1 ton, 2 tons, 2½ tons, and 3 tons, in such a way, as to imitate as nearly as possible actual conditions of working, without being subjected to the delays so frequently experienced in discharging ore or general cargoes.*

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* In calculating the weight of cargo discharged per hour, allowance was made for the 9-cwt. empty buckets and the 2-cwt. hoisting chain balls, as follows; (9 − 2) cwts. = ·35 ton. Now, this weight was lifted only 10 feet during the return journey with the empty bucket, and = (·35 × 10) ft.-tons of work. But each ton of paying load is lifted 30 feet = 30 ft.-tons of work. Hence, for every 30 ft.-tons of general cargo work, there would have to be added in this test, 3·5 ft.-tons for the empty bucket minus the chain ball. Or, $\left(\cdot 35 \times \frac{10}{30}\right) = \cdot 116$ ton, as shown in the two following tables.

Curve Diagrams 1 and 2.—The principal curves to which attention may be directed are the speed lines, **S**, which show that both cranes are nearly alike as regards speed at all loads—varying from 260 to 165 feet per minute in the case of the hydraulic crane, and from 239 to 165 feet per minute in the electric crane, at loads of ½ and 3½ tons respectively.

The efficiency line, **E**, for the hydraulic crane rises in a straight line from 9½ per cent. to 66 per cent. for loads of ½ and 3½ tons respectively. The efficiency line, **E**, of the electric crane rises in a full, well-rounded curve, showing at

½ ton	1 ton	2 tons	2½ tons	3 tons	3½ tons
efficiencies of					
31%	49·34%	73·14%	77·1%	75·3%	70%

The other lines, **C**, show the cost of power used in raising 30 feet, slewing 100 degrees, and discharging into waggons 1,000 tons of cargo. It will be seen, that the cost line for the hydraulic crane rises in a steep curve at the light loads, while that for the electric crane is a straight line nearly parallel with the abscissæ, showing how the cost increases in the case of the former when working at light loads, while the cost is practically constant at any load with the electric crane.*

* I have added to these two diagrams E_C lines. In the case of the hydraulic crane this dotted curve, E_C, indicates the percentage efficiency as found by the pressure gauges attached directly to the hydraulic cylinder. This curve, therefore, represents the ratio of the power got out at the crane hook (when lifting loads 30 feet in height), to the power put into the cylinder during that time. It clearly shows the effect of the loss of pressure between the hydrant on the main supply pipe and the ram cylinder. It also shows by comparison the efficiency of the plant between the crane hook and the cylinder ram with that between the same hook and the power put into the mains at 750 lbs. per square inch.

In the case of the electric crane, the dotted curve, E_C, is derived from special sets of ten observations for each load, taken on March 20th, 1903. The times of simply lifting loads 30 feet were accurately noted, as well as the readings on the crane volt and ampere meters during these respective lifts, quite independently of the meters at the power-house. This curve, therefore, represents the best net combined electrical and mechanical efficiency of the crane, whilst merely lifting different loads. It indicates the ratio of the power got out at the crane hook, to the power put into the lifting motor, or the crane efficiency

Note.—Students may refer here to *Engineering* of June 19 and July 3, 1903, for reports upon two papers, "Economical Speeds of Cranes," and "Hydraulic *versus* Electric Cranes for Docks," read at the London Engineering Conference.

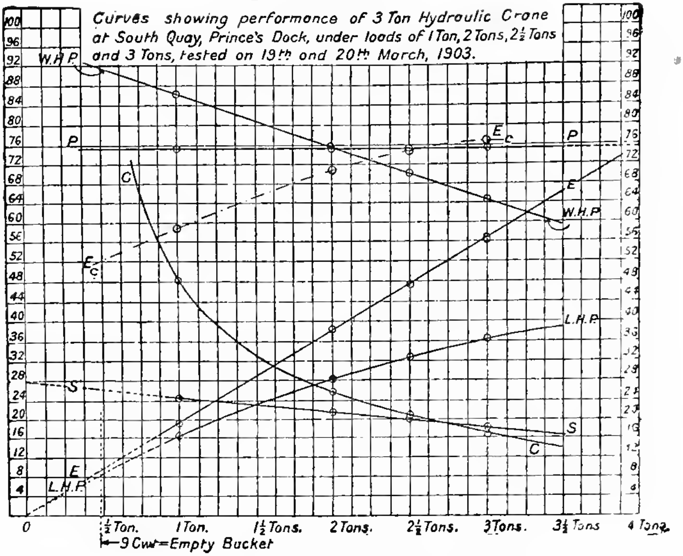
TABLE I.—Results of Tests of 3-Ton Hydraulic Crane Worked as if Discharging General Cargo.

Loads (W).	1 Ton.	2 Tons.	$2\frac{1}{2}$ Tons.	3 Tons.
Duration of trial, hours,	2	2	2	2
Pressure in mains, lbs. per sq. in.,	750	750	750	750
Pressure on ram—				
Hoisting, . . ,,	240	400	480	560
Lowering, . . ,,	140	300	380	460
Slewing to windward, ,,	500	500	500	500
,, leeward, ,,	350	350	350	350
1. Number of Cycles per Hour, ...	56	54	52	38
Time of lifting W 30 feet (T), . . seconds,	7·5	8·57	9·25	10
Speed of hoisting loads $(S)=\frac{30\times 60}{T}$ ft. per min.,	240	210	194·6	180
H.P. taken out of crane $(L.H.P.)=W\times S\div 33{,}000$,	16·29	28·51	33	36·65
Horse-power, by water $(W.H.P.)=\frac{95\times 750\times S}{33{,}000\times 6}$,	86·36	75·56	70	64·77
2. Mechanical Efficiency (*Hoisting only*), $\%=100\times\frac{L.H.P.}{W.H.P.}$, .	18·87	37·73	47·14	56·6
Cargo discharged per hour (allowing for empty bucket $·35\times\frac{10}{30}$ ton), tons,	$(1·116\times 56)$ 62·5	$(2·116\times 54)$ 114	$(2·616\times 52)$ 136	$(3·116\times 38)$ 118·5
Water used per cycle—				
Hoisting 30 feet, cub. ft.,	3·3	...	...	...
Slewing 100°, ,,	·57	...	...	...
Hoisting 10 feet, ,,	1·1	...	...	...
Slewing 100°, ,,	·57	...	...	...
Slip, 5 per cent., ,,	·28	..	...	...
Total cubic feet, .	5·82	5·82	5·82	5·82
Water used per hour, cub. ft.,	326·5	314·5	303	221·5
Cargo discharged per hour, tons,	62·5	114	136	118·5
Water used per 1,000 tons cargo, . . cub. ft.,	5,224	2,759	2,228	1,869
3. Cost of Hydraulic Power used in discharging 1,000 tons general cargo, at 110·45 pence per 1,000 cubic feet of water at 750 lbs. per square inch (C), . .	48/	25/5	20/6	17/2

TABLE II.—RESULTS OF TESTS OF 3-TON ELECTRIC CRANE WORKED AS IF DISCHARGING GENERAL CARGO.

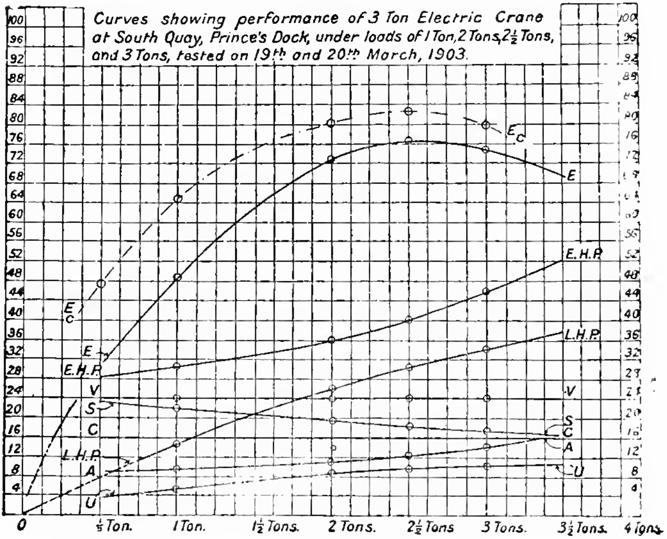
LOADS (W)	1 Ton.	2 Tons.	$2\frac{1}{2}$ Tons.	3 Tons.
Duration of trial, hours,	2	2	2	$\frac{1}{4}$
E.M.F. at motor terminals, . . volts,	240	245	240	245
Current when hoisting loads, amps.,	95	110	125	140
Current when lifting empty bucket, . amps.,	60	60	60	60
Current slewing to windward, amps.,	15	35	35	40
Current slewing to leeward, . . amps.,	15	15	15	15
1. NUMBER OF CYCLES PER HOUR, . . .	60	53	48	43
Work done by hoisting W 30 feet, . . foot-lbs.,	67,200	134,400	168,000	201,600
Time of lifting W 30 feet (T), . seconds,	8·1	9·25	9·85	10·6
Speed of hoisting loads, $S = \frac{30 \times 60}{T}$ ft. per min.,	222·2	194·5	182·7	170
H.P. taken out of crane (L.H.P.) $= W \times S \div 33{,}000$.	15·08	26·42	31	34·62
Horse-power expended, (E.H.P.) $= \frac{\text{volts} \times \text{amps.}}{746}$.	30·56	36·12	40·21	46
2. COMBINED MECHANICAL AND ELECTRICAL EFFICIENCY, °/。 $= 100 \times \frac{L.H.P.}{E.H.P.}$ (*Hoisting only*), . .	49·34	73·14	77·1	75·3
Cargo discharged per hour (allowing for empty bucket ·35 × $\frac{19}{20}$ ton), tons,	$(1·116 \times 60)$ 67	$(2·116 \times 53)$ 112	$(2·616 \times 48)$ 125·5	$(3·116 \times 43)$ 134
Units used per hour at various loads, . . .	5	8·5	9·5	10·4
Units required for 1,000 tons cargo (U), . .	74·6	75.8	75·7	77·6
3. COST OF ELECTRIC POWER used in discharging 1,000 tons general cargo, at 2·558 pence per unit (C).	15/11	16/2	$16/1\frac{1}{2}$	$16/6\frac{1}{2}$

DIAGRAM Nº 1.



Curves showing performance of 3 Ton Hydraulic Crane at South Quay, Prince's Dock, under loads of 1 Ton, 2 Tons, 2½ Tons and 3 Tons, tested on 19th and 20th March, 1903.

DIAGRAM Nº 2.



Curves showing performance of 3 Ton Electric Crane at South Quay, Prince's Dock, under loads of 1 Ton, 2 Tons, 2½ Tons, and 3 Tons, tested on 19th and 20th March, 1903.

EXPLANATION OF EFFICIENCY CURVES.

Hydraulic Crane.

P. Pressure in mains during tests.
Each division = 40 lbs. per square inch.

S. Speed of hoisting in feet per minute.
Each division = 40 feet per minute.

W.H.P. Horse-power put into crane by water at 750 lbs. per square inch, hoisting only.
Each division = 4 H.P.

L.H.P. Horse-power taken out of crane in hoisting only.

E. $^\circ/_\circ$ Mechanical efficiency $= \frac{\text{L.H.P.}}{\text{W.H.P}} \times 100$ (*hoisting only*).
Each division = 4 per cent.

$\mathbf{E_C}$. $^\circ/_\circ$ Efficiency $= \frac{\text{Power got out at crane hook}}{\text{Power put into hydraulic ram}}$, when *merely* lifting different loads 30 feet during special trials.
Each division = 4 per cent.

C. Cost of power used in raising 30 feet, slewing 100°, and discharging into waggons 1,000 tons of cargo.
Each division = 4 shillings.

Electric Crane.

V. Volts while hoisting at speed (**S**).
A. Amperes ,, ,,
Each division = 40 volts and also = 40 amperes.

S. Speed of hoisting in feet per minute.
Each division = 40 feet per minute.

E.H.P. Horse-power delivered to hoisting motor.
Each division = 4 H.P.

L.H.P. Horse-power taken out of crane in hoisting only.

E. $^\circ/_\circ$ Electrical and mechanical efficiency $= \frac{\text{L.H.P.}}{\text{E.H.P.}} \times 100.$
Each division = 4 per cent (*when hoisting only*).

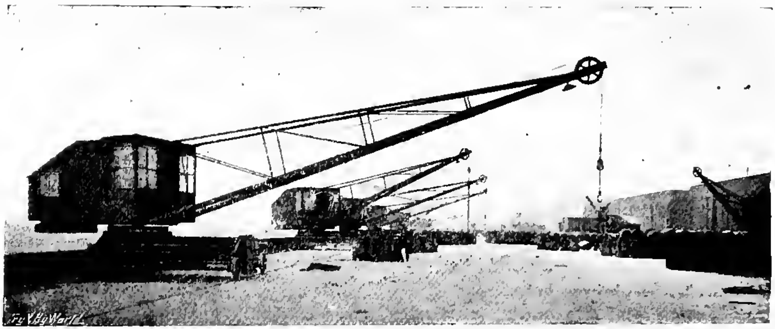
$\mathbf{E_C}$. $^\circ/_\circ$ Crane efficiency $= \frac{\text{Power got out at crane hook}}{\text{Power put into motor}}$ when *merely* lifting different loads 30 feet during special trials.
Each division = 4 per cent.

C. Cost of power used in raising 30 feet, slewing 100°, and discharging into waggons 1,000 tons of cargo.
Each division = 4 shillings.

U. Units used per hour at various loads.
Each division = 4 units.

Electric Cranes for Manchester Ship Canal.—As one more instance of the rapid progress which the application of electricity to cranes is making, we here show two views of those recently made and fitted by George Russell & Co., of Mother well, for the Manchester Ship Canal Warehouse Co. There are twenty two cranes placed on the roof of the warehouse. Of these, twelve are portable cranes on the dock side of the warehouse for lifting 30 cwts., with a radius of 42 feet. The remaining ten are fixed cranes on the land side of the warehouse, three of which lift 30 cwts., and seven 10 cwts., both having a radius of 16 feet.

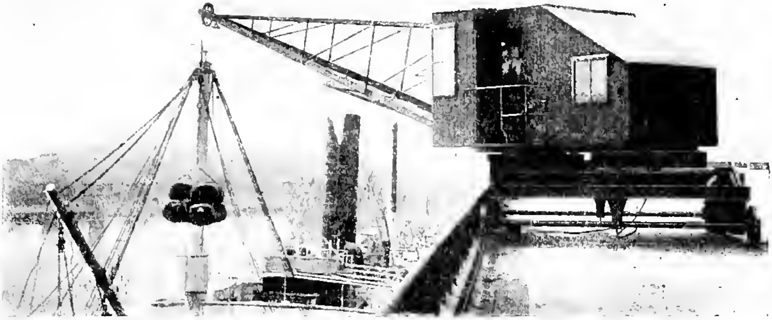
The electrical motors and controllers were supplied by the British Thomson Houston Co., Rugby. The hoisting motors for the 30-cwt. cranes are rated at 28 H.P. when their armatures have a speed of 355 revs. per minute, and are supplied at 500 volts. They are geared direct to the lifting barrels by single reduction machine-cut spur wheels. The barrel of each of these cranes is 24-inches diameter, and the lift is effected by a single wire rope. The speed of lifting was specified to be 200 feet per minute, with a full lift of 75 feet. A strap brake operates upon a brake wheel, keyed upon the barrel shaft. Under normal conditions the brake bears upon its brake wheel through the action of a lever weight, just as in the previously described Clyde Trust Electric Crane. But, in this instance, the weight is raised by an electro-magnet in series with the lifting motor,



FIXED AND MOVABLE ELECTRIC CRANES ON THE MANCHESTER CANAL WAREHOUSE.
(Made by George Russell & Co., Ltd., Motherwell, N.B.)

so that whenever current is switched on to the motor the weight is lifted and the brake released. In the event of the current failing through any cause the pull of the magnet ceases, and the weight automatically applies its brake to prevent the load from falling. The brake may be also released by a foot tread, so that the load may then be lowered without using any current.*

From a final test of one of the portable cranes it was found that whilst lifting four bales of cotton at a time, having an average weight of 3,136 lbs., through 43½ feet, that the time was 11·5 seconds (or 227 feet per minute), current 38·5 amperes at 510 volts; consequently, the efficiency under these circumstances reached 82 per cent.



MOVABLE ELECTRIC CRANE UNLOADING A STEAMER AT THE MANCHESTER CANAL.

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* For a fuller description of the details of these cranes students may refer to the March and April numbers of the leading British electrical and mechanical journals, such as *The Tramway and Railway World*, April 1903, to which paper I am indebted for the two views here reproduced.

LECTURE II.—QUESTIONS.

1. Explain why the force necessary to drive a machine does not vary in exact proportion with the load.

2. With a pair of three sheaved blocks it is found by experiment that a weight of 40 lbs. can be raised by a force of 10 lbs., and a weight of 200 lbs. by a force of 40 lbs. Find the general relation between P and W, and also the efficiency when raising 100 lbs. *Ans.* $P = \frac{3}{16}W + \frac{5}{2}$; ·784 or 78·4 per cent.

3. With a screw jack it is found that a force of 47·5 lbs. must be applied at the end of the handle to lift 1·5 tons, and a force of 85 lbs. to lift 3 tons. Find what force will be required to raise 2 tons. *Ans.* 60 lbs.

4. If the length of the handle in the above example be 2 feet and the pitch of the screw $\frac{3}{4}$ inch, find the efficiency in each case. *Ans.* 35·3, 39·5, and 37·3 per cent.

5. Find the "*friction pressure*" of a steam engine which requires a mean effective pressure of 24 lbs. per square inch to drive it at full load, 20 lbs. being taken up in overcoming the load. At three-quarters load a mean effective pressure of 18·5 lbs. is required, of which 15 lbs. is similarly taken up. *Ans.* 2 lbs. per square inch.

6. It is found that a force of 2 lbs. must be applied to the handle of a crane in order to wind up the rope when no weight is attached and of 80 lbs. when lifting a weight of 10 cwts. If the velocity ratio be 20, find the efficiency in this last case and also when the force at the handles is lessened so as just to allow the weight to descend. *Ans.* 70 per cent.; 57 per cent.

7. A machine is concealed from sight except that there are two vertical ropes; when one of these is pulled downwards the other rises. If the falling of a weight, A, on one, causes a weight, B, on the other to be steadily lifted, first when A is 100 lbs. and B is 3,250 lbs., second when A is 50 lbs. and B is 1,170 lbs., what is the probable value of A when B is 2,000 lbs.? How would you find the efficiency of this lifting machine in these three cases? What new measurement must be made?
Ans. 69·95 lbs. (B. of E. Adv., 1900.)

8. A machine is concealed from sight, except that there are two vertical ropes; when one of these is pulled downwards the other rises. If the falling of a weight, A, on one, causes a weight, B, on the other to be steadily lifted, first when A is 12 lbs. and B is 700 lbs., second when A is 7·6 lbs. and B is 300 lbs. What is A likely to be when B is 520 lbs.? If B rises 1 inch when A falls 70 inches: what is the efficiency of this lifting machine in each of the three cases? (B. of E. Adv., 1901.)

9. In a lifting machine an effort of 26·6 lbs. just raises a load of 2,260 lbs., what is the mechanical advantage? If the efficiency is 0·755, what is the velocity ratio? If on this same machine an effort of 11·8 lbs. raises a load of 580 lbs., what is now the efficiency? (B. of E. Adv. & H., Part I., 1902.)

10. If the diameter of the 3-ton lifting ram of the hydraulic crane described in this lecture is 11 inches, and its stroke be 5 feet when the chain hook rises 30 feet, and if the inside diameter of the water service pipe from the power-house to the crane dock hydrant opposite this crane is 7 inches; find what length and weight of a column of water in the service pipe is used at each complete 5-foot stroke of the lifting ram.

11. Referring to the previous question, if a 3-ton load be lifted 30 feet high in 10 seconds, and if the lengths of 7-inch service piping between the crane hydrants and the power-house accumulator be 500 and 1,000 feet respectively for two different cranes, what will be the loss in pressure in lbs. per square inch between the power-house and each of these two hydrants, when accumulator applies a steady pressure of 750 lbs. per square inch at the power-house, and if the coefficient of friction between the water and its service pipe (or the "friction factor") is ·02 at the velocity under these circumstances? Also, calculate (*a*) the ft.-lbs. of work; (*b*) the watt-hours of work lost through the friction in the pipe at each lift of 3 tons; (*c*) the ft.-lbs. per second; (*d*) the watts or the lost power in the water service pipe during the lift.*

12. Referring to Table I. of tests and the curves for the 3-ton hydraulic crane in this Lecture, how do you account for the differences in the loss of the pressures per square inch between the water in the mains and the pressures on the ram whilst hoisting different loads? What would be the calculated corresponding loss of pressure in each case between the power-house and the crane hydrant if the distance of the main supply pipe was 1,000 feet long and the times of lifting 1, 2, $2\frac{1}{2}$, and 3 tons as stated in Table I.?

13. Check by calculations the several values for W.H.P., L.H.P., E.H.P., E., and C., in the case of the hydraulic and electric cranes described in this Lecture, and draw their curves upon squared paper neatly to a scale at least double the size of that given in the text.

14. Referring to Tables I. and II., &c., in this Lecture, calculate and compare the loss of power in ft.-lbs. per second (or in watts) for the hydraulic and the electric mains whilst lifting 1, 2, $2\frac{1}{2}$, and 3 tons, and show clearly which system of distribution of power is more efficient. How would you propose to render the distribution of power from the power-house to the electric crane more economical?

N.B.—*See Appendices B and C for other questions and answers.*

LECTURE II.—A.M.INST.C.E. EXAM. QUESTIONS.

1. What is meant by the efficiency of a machine? How would you measure the efficiency (1) of a screw-jack, (2) of a steam pump, (3) of a turbine? (I.C.E., *Oct.*, 1897.)

2. An hydraulic company charges 15d. per 1,000 gallons of water at 820 lbs. per square inch; what is the cost per H.P. hour? (I.C.E., *Oct.*, 1898.)

3. The saddle of a lathe weighs 5 cwts., and it is moved along the lathe bed by a rack and pinion arrangement. What force applied at the end of a handle 10 inches long will be just capable of moving the saddle, supposing the pinion to have twelve teeth of 1¼ inch pitch, the coefficient of friction between the saddle and lathe bed to be 0·13, and if 35 per cent. of the work done in working the rack and pinion is expended in overcoming friction. (I.C.E., *Feb.*, 1902.)

4. An experiment on a windlass to determine the effort required to lift different loads, gave the following results:—

Load (lbs.)	0	28	56	84	112	140	168	196	224
Effort (lbs.)	1·5	2·75	4	5·2	6·4	7·6	8·8	9·9	11

The velocity ratio of the machine was 52. Plot the results on squared paper, showing the effort curve on a load base, and draw also the friction and efficiency curves, finding at least five points in each. Find the effort for a load of 210 lbs., and find the mechanical advantage and mechanical efficiency of this load. (I.C.E., *Feb.*, 1902.)

N.B.—*See Appendices B and C for other questions and answers.*

LECTURE III.

HYDRAULIC APPLIANCES IN GAS WORKS.

Contents.—Labour-Saving Appliances in Modern Gas Works—Pumping Engines and Accumulator—Example I.—Differential Accumulator—Brown's Steam Accumulator—Small Hydraulic Accumulator Plant—Arrol-Foulis' Gas Retort Charging Machine—Foulis' Withdrawing Machine for Gas Retorts—Results of Working—Questions.

Labour-Saving Appliances in Modern Gas Works.*—In no industry have the conditions of labour undergone more radical and rapid changes than in large modern gasworks. Until recently, the coal was emptied by hand from railway waggons or from carts at the most convenient position for the retort benches. If the lumps of coal were too large, they were broken up by manual labour, and the retorts were charged, as well as discharged, by hand. The surplus coke, beyond what was required for the furnaces, was quenched, wheeled into a yard, and there screened and filled into carts or waggons, all by hand labour.

In modernised works, the coal is usually delivered direct from the railway truck into the hopper of the mechanically-driven coal breaker. After passing through this machine, it is raised by an elevator to a large hopper, which is so fixed that the coal can be automatically delivered into the hopper of any of the charging machines. The charging of the retorts is then performed by means of a hydraulic charging machine more evenly and otherwise better than by hand. After carbonisation, the coke is withdrawn from the retorts by a hydraulic drawing machine. These two operations now entail a minimum of labour and inconvenience from heat to the attendants. The surplus coke, guided in its descent by shoots, falls into a bogie underneath the floor, and is run out to the yard by small locomotives. In some instances conveyors, which are placed underneath the stage floor, carry the coke to circular revolving screens, whence it is delivered into large storage hoppers, railway waggons, or bags for small consumers.

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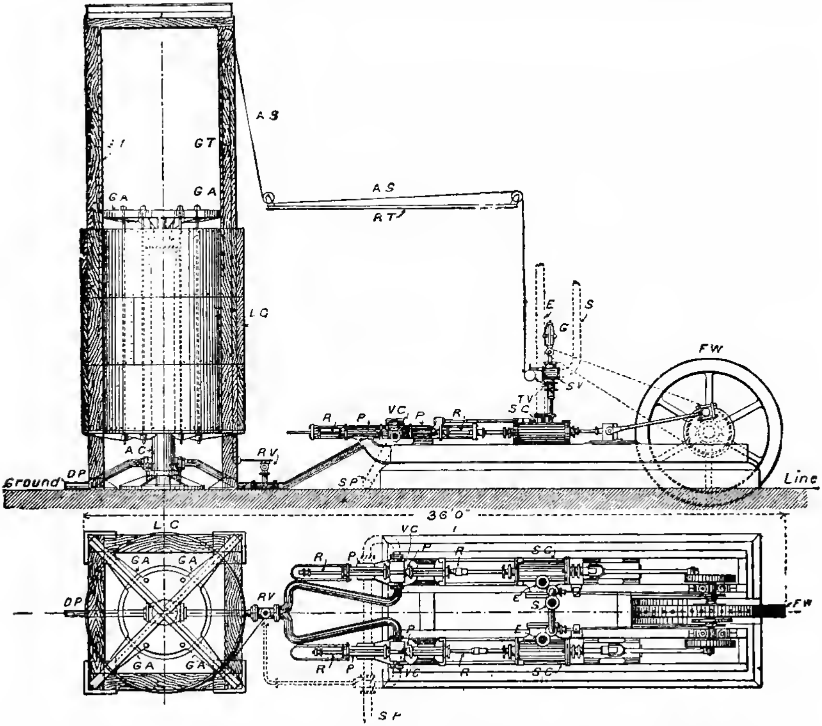
* I am indebted to Mr. Andrew S. Biggart, of Sir William Arrol & Co., Ltd., Glasgow, for the drawings from which the three electros of the General Arrangement, Charging, and Drawing Machines were made. I am also indebted for the information contained in Mr. Biggart's paper which he read before the 1895 Glasgow meeting of the Institution of Mechanical Engineers, as well as to Mr. Foulis, the General Manager of the Glasgow Corporation Gas Trust for showing me the whole of the plant in operation.

The old methods involved continuous and repeated operations performed by hand; while the new are such that no hand labour is employed in dealing with the coal from the time it leaves the railway truck until it arrives there again in the form of coke. Hand labour is thus entirely superseded by mechanical power, to the great advantage of both the labourer and the employer. The number of men required in the retort house under the new system is less than half that required under the old method. The saving which this represents, after allowing for maintenance of plant and interest on additional capital, will average about one shilling per ton.

Pumping Engines and Accumulator.—The general arrangement of this portion of the plant is illustrated by the following side elevation and plan. The steam engine and pumps are situated either in the main engine room of the gasworks or in a convenient house not far removed from the steam boilers ; while the accumulator may be placed in an annex, where there is plenty of head room for the guide timbers. From what has already been said about the construction and action of the Armstrong accumulator in the preceding Lecture, and by carefully comparing the present figures with the index to parts (which has been tabulated in nearly the exact sequence of the operations) the student will have no difficulty in understanding the generating plant which serves to supply hydraulic power to the charging and drawing machines in the retort houses.

The steam engine drives the rams R of the pumps P direct from a back extension of the piston-rods. From the pumps the water is forced into the accumulator cylinder A C at such a pressure that, acting on the accumulator ram, it is capable of lifting the heavy dead-weight bolted to the upper end. This weight is provided with guide arms G A, bearing upon planed iron rails fixed to the vertical guide timbers G T. When the guide arms have reached a certain height, the one at the right hand begins to lift a weight attached to a chain on the weighted lever of the throttle valve T V. During the remainder of the ascent of the accumulator the automatic starting chain A S is slackened until the throttle valve is closed, and the engine is stopped before the guide arms reach the cross beam, which connects the upper ends of the guide timbers. Should the chain stick, or anything fail about this automatic system of stopping the engine, then a second chain, which connects the weight at the end of the relief valve R V to the counterpoise ball



General Arrangement of Pumping Engines and Accumulator for Working Gas Retort Charging and Withdrawing Machines.

Index to Parts.

S	for	Steam Pipe.	V C	for	Valve Chest for Pumps.
S V	,,	Stop Valve.	R	,,	Rams.
T V	,,	Throttle Valve.	P	,,	Pumps.
A S	,,	Automatic Starting and Stopping Apparatus.	R V	,,	Relief Valve.
			A C	,,	Accumulator Cylinder.
S C	,,	Steam Cylinder.	D P	,,	Delivery Pipe.
E	,,	Exhaust Pipe.	G A	,,	Guide Arms.
G	,,	Governor.	G T	,,	Guide Timbers.
F W	,,	Flywheel.	L C	,,	Load Casing.
S P	,,	Suction Pipe.	R T	,,	Roof Tie-Rods.

attached to the chain A S, is tightened up just before the guide arms reach the uppermost limit of their travel.* This action releases the downward pressure on the relief valve, and permits sufficient water to escape to prevent any further elevation of the accumulator ram. The governor G now prevents the engine from racing. When water is used by any of the hydraulic machines it flows to them from the accumulator cylinder through the delivery pipe D P. This allows the accumulator ram and weight to sink until the ball attached to A S re-asserts its pull on the throttle valve, and, automatically opening it, starts the engine.

It is evident that if a set of engines and pumps were devoted to the direct supply of high-pressure water to several machines, their output would have to be equal to the greatest requirements of the plant at any instant; but by the introduction of the accumulator and its automatic starting and stopping gear we have a simple means of ensuring a constant supply of water at the desired pressure with a smaller engine.

An accumulator, therefore, performs several very important functions in a most efficient manner:—

(1) It acts as a reservoir for the storage of energy.

(2) It acts as a regulator of pressure.

(3) It acts like a flywheel in taking up and giving out power in direct sympathy with the immediate wants of supply and demand.

(4) It acts as an elastic buffer, and prevents the breakage of joints, &c.

(5) It automatically controls the motive power.

The efficiency of an accumulator has been proved to be as high as 98 per cent., only 1 per cent. being lost through friction in charging, and 1 per cent. in discharging it, as tested by pressure gauges on the supply and discharge pipes. Its total store of energy is, however, comparatively small, since it is only equal to the potential energy of the weight raised. Hence, if W lbs. be the total weight raised in the accumulator, and H feet the difference of height between its highest and lowest positions, we have:—

$$\left.\begin{array}{l}\textbf{Energy Stored in}\\ \textbf{Accumulator}\end{array}\right\} = \text{W H ft.-lbs.} = \frac{\text{W H}}{33{,}000 \times 60}\ \text{H.P.-hours.}\dagger \qquad \text{(I)}$$

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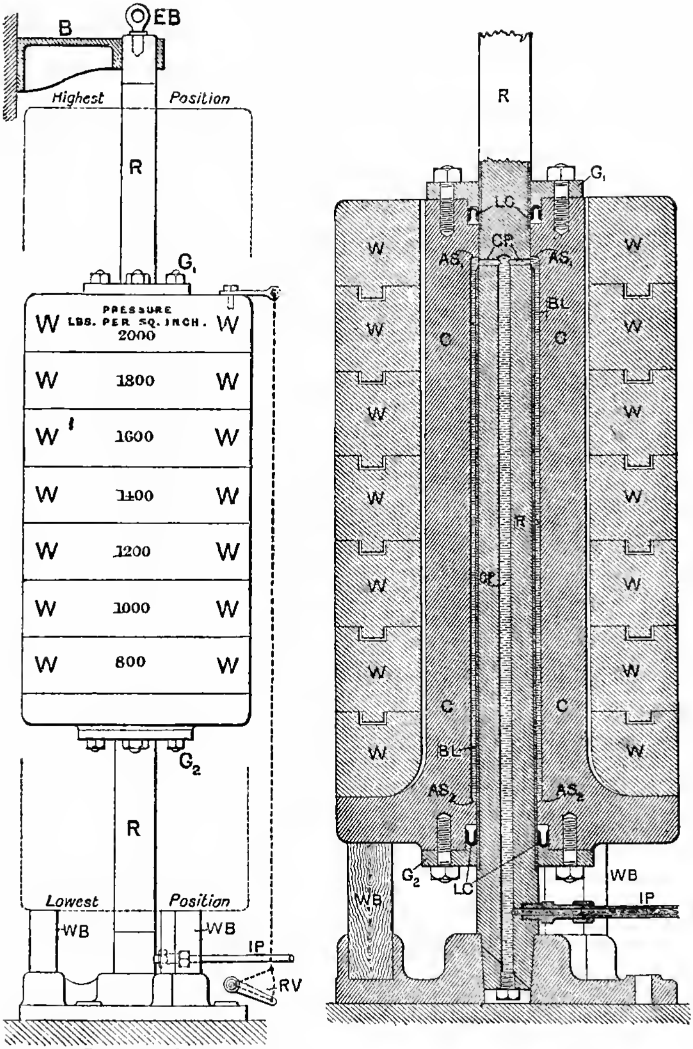
* Another arrangement is to pass this second vertical chain through a hole in a projecting plate fixed to the right-hand guide arm G A, and then attach its upper end to the top cross beam. If the accumulator should rise too high, then the plate catches an enlarged portion of the chain and opens the relief valves.

† One horse-power = 33,000 ft.-lbs. of work per minute, hence, 1 horse-power hour is 33,000 × 60 or 1,980,000 ft.-lbs.



ELEVATION.

VERTICAL SECTION.

TWEDDELL'S DIFFERENTIAL ACCUMULATOR.

Example I.—The accumulators used in connection with the hydraulic power supply in Glasgow are 18 inches in diameter, and have a free lift of 23 feet. The total load on each is 127 tons. Find the pressure of the water and the maximum energy stored in each accumulator, neglecting friction.

Answer.—

Pressure of water $= p = \dfrac{W}{A}.$

„ „ $= \dfrac{127 \times 2{,}240}{\cdot 7854 \times 18 \times 18} = 1{,}120$ **lbs. per sq. in.**

Energy stored $= WH = 127 \times 2{,}240 \times 23.$

„ $= 6{,}543{,}000$ ft.-lbs., or **3·3 H.P.-hours.**

Index to Parts.

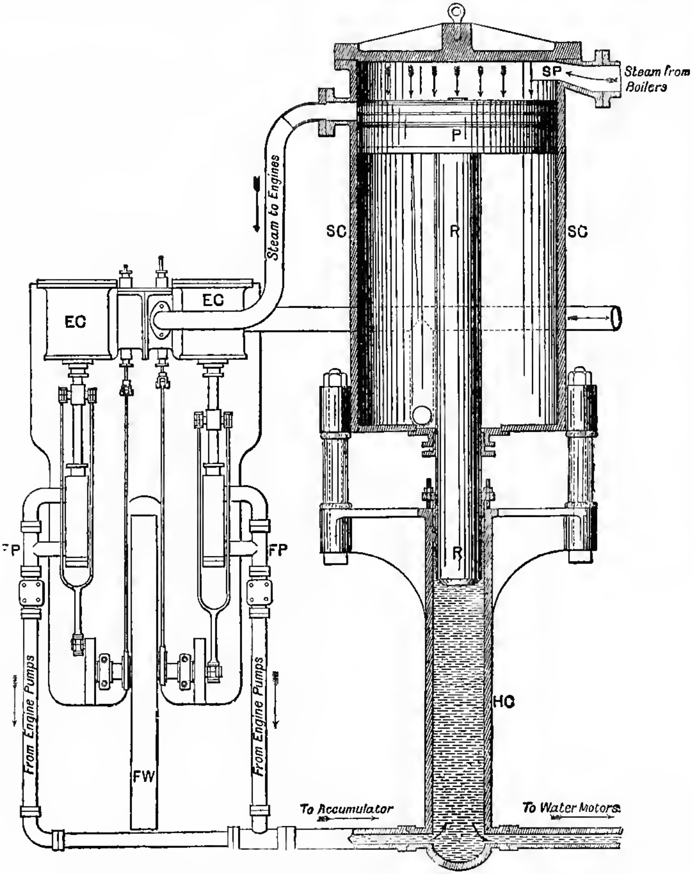
I P	for	Inlet Pipe from pump.
R	,,	Ram of accumulator.
B L	,,	Brass Liner on lower part of R.
C P	,,	Central and Cross Passages for water.
C	,,	Cylinder.
A S_1, A S_2	,,	Annular Spaces from top and bottom of cylinder.
G_1, G_2	,,	Glands (top and bottom).
L C	,,	Leather Cup Packings.
W	,,	Weights.
W B	,,	Wooden Blocks.
B	,,	Bracket (at top).
E B	,,	End Bearing.

It will be seen from this example how small the total store of energy is, and that accumulators would be quite useless to maintain the supply for any length of time. One of the real advantages of the accumulator arises from the fact that we may use its energy for a very short time at a high rate. For instance, although the accumulator in the above example is only capable of maintaining 3·3 H.P. for a whole hour, it could exert 19·8 H.P. for ten minutes, or 198 H.P. for one minute.

Differential Accumulator.—Although it has only been found necessary to employ a water pressure of 400 lbs. per square inch in the charging and drawing machines for gas retorts, which we will describe later on, and, further, since it is advisable with them to have a considerable volume of water in the accumulators when many machines have to be worked simultaneously, yet it may not be out of place to describe here the differential accumulator designed by Mr. Tweddell for his smaller kinds of hydraulic tools, since there may be cases in which space and compactness are of considerable importance. From the foregoing elevation and enlarged vertical section it will be seen that the ram R consists of a vertical fixed shaft secured at the top by a bracket B, and at the bottom by a footstep. The lower half of this shaft is of larger diameter than its upper half, a brass liner B L being shrunk on the former part. Moreover, this lower portion of the ram has a central passage C P drilled axially along it, with a cross passage just above the upper end of the brass liner. Through these passages water is admitted from the inlet pipe I P, which is connected directly to the force pumps. This water finds its way into an annular space A S_1, A S_2, which is the clearance between the outside of the brass liner and the inner bore of the heavy press or cylinder C. Surrounding the outside of the cylinder are placed a number of cast-iron or lead weights W which fit into each other, and form the dead load along with the weight of the cylinder. At the top and the bottom of the cylinder there are suitable glands G_1 and G_2 containing the usual leather cup packings L C. When the machine is idle the bottom flange of the cylinder rests upon wooden beams W B. It will now be readily understood that the effective area of the ram is *only* the difference between the cross areas of the brass liner B L and the upper part of the ram R, instead of the whole area of the ram as in the previous case. Hence, a very great pressure may be obtained from a small weight. For example, should the annular area representing the difference in size between the brass liner and the upper part of the iron ram be 5 square inches, and the total weight of the cylinder and its surrounding cast-iron blocks be 2,000 lbs., then, neglecting the friction at the glands, the pressure would be 2,000 ÷ 5, or 400 lbs. per square inch. This accumulator will store up an equal amount of energy, as in the previous case, if the dead weight and height of the lift are the same since their stored energy depends directly upon W × H. The volume of water contained in the accumulator will, however, be comparatively small, and hence it will fall more quickly for a certain amount of water used by the hydraulic machines which it drives. As will be seen from the left-hand figure, a relief valve R V is worked by a chain connecting an outstanding arm on the uppermost weight to the end of the lever; also any desired pressure up to 2,000 lbs. per square inch, or more, may easily be obtained from this accumulator with a comparatively small dead load and space.

Brown's Steam Accumulator.—Another very simple form of accumulator, which has proved very effective both for land purposes and on board ship, is that designed and made by Mr. A.

Betts Brown, of the Rosebank Iron Works, Edinburgh. It consists of a steam cylinder S C fitted with a piston P and a piston-rod or ram R. Steam is supplied direct to this cylinder from the boiler and presses on the piston P in opposition to the



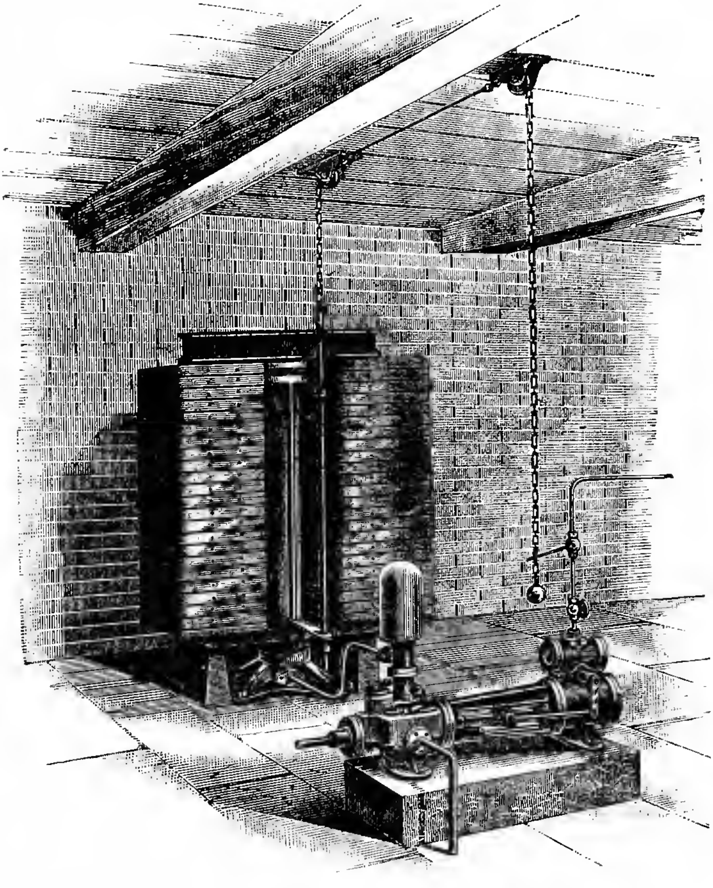
BROWN'S STEAM ACCUMULATOR OR COMPENSATED STEAM PUMP.

INDEX TO PARTS.

S P for Steam Pipe from Boilers.
S C ,, Steam Cylinder.
P ,, Piston working in S C.
R ,, Ram attached to P.

H C for Hydraulic Cylinder.
E C ,, Engine Cylinders.
F P ,, Force Pumps.
E ,, Exhaust Pipe.

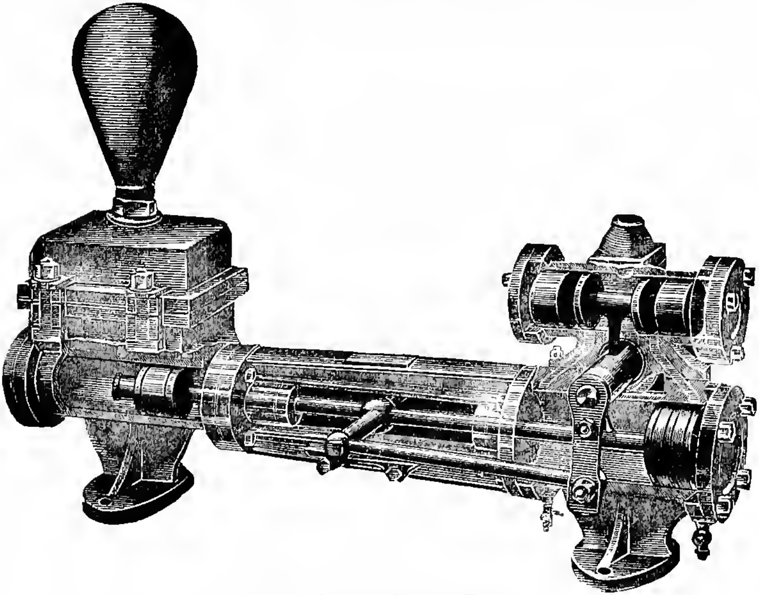
water forced into the hydraulic cylinder H O by the force pumps F P, which are worked by a pair of engines. An exhaust pipe E carries away the exhaust steam from the engine cylinders E C and the bottom of the large steam accumulator cylinder S C.



Small Hydraulic Accumulator Plant.

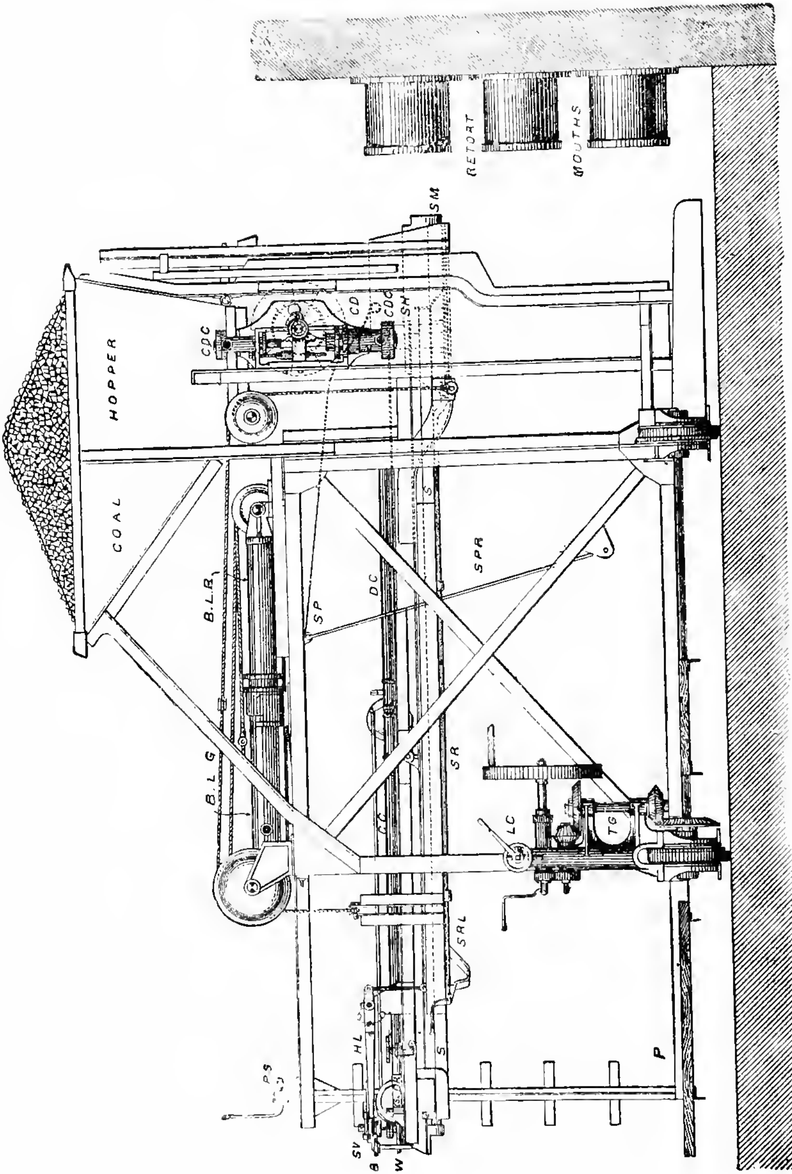
Suppose that the piston P is at the bottom of its cylinder, then the boiler steam not only fills the portion above the piston but passes on to the engine cylinders and therefore works the pumps and forces up the ram and piston to the top of their stroke, when the piston gradually closes the steam passage leading to the engines until they stop. Should any of the hydraulic machines be now put into action the water flows to them from the hydraulic cylinder, the ram and piston descend and the engines are again set into motion to keep up the demand and again close the engine steam pipe. With a steam cylinder of 36 inches diameter and a ram of $9\frac{1}{4}$ inches (or, ratio of areas 15 to 1) Mr. Brown is able with about 50 lbs. steam pressure per square inch to maintain a pressure of 750 lbs. per square inch in the hydraulic mains leading to water motors, steering, stopping, and starting gear, or, in the case of a gas works, to the charging and drawing machines.

Small Hydraulic Accumulator Plant.—Should the gas works be a small one, then a much less expensive accumulator plant may be employed, such as that illustrated. This figure is sufficiently self-descriptive after what has been said about the larger plant.



DIRECT ACTING STEAM PUMP.

The small pumping engine, or donkey pump, which supplies water of the desired pressure to the accumulator is also shown in a perspective view. Steam is admitted to the steam



ARROL-FOULIS GAS RETORT CHARGING MACHINE.

cylinder by piston valves, actuated by a connecting-rod and lever, while the double-acting pump is fitted with a solid piston plunger and an air vessel.

Arrol-Foulis Gas Retort Charging Machine.—In replacing hand labour for the charging gas retorts by a hydraulic machine several important requirements have to be effected:—

(1) The machine should be easily moved parallel to the retort bench at the proper distance therefrom.

(2) It must be capable of being connected with the hydraulic pressure pipe at every position.

(3) The shoot mouth should be easily raised or lowered to suit the highest or lowest retort.

(4) The shoot mouth should be easily entered into the mouth of the retort before filling it with coal.

(5) The coal hopper should contain at least a sufficient quantity of coal to charge a complete set of retorts without requiring to be refilled.

(6) The exact quantity of coal required for the charge of each retort should be easily regulated, and the regulator should be capable of dealing with all the sizes of coal in use.

(7) The charge should be laid evenly and of the desired depth from back to front without any special effort on the part of the attendant, and without loss of time.

(8) The platform of the attendant should be placed in the most convenient position for actuating all the motions, and at the same time protecting him from the heat emanating from the retorts.

(9) The whole machine should require a minimum of attention and repair. It should also be able to withstand the rough usage of retort house workmen, as well as the hydraulic pressure, without undue leakage or giving way in any of its vital parts.

INDEX TO PARTS.

P for Platform (for attendant).
T G ,, Travelling Gear.
P S ,, Pressure Swivel.
L C ,, Lifting Cock.
B L C ,, Beam Lifting Cylinder.
B L R ,, Beam Lifting Ram.
S R L ,, Shoot Rod Lever.
S R ,, Shoot Rod.
S M ,, Shoot Mouth.
H L ,, Hand Lever.
C D C ,, Coal Drum Cylinders.

C D for Coal Drum.
S V ,, Slide Valve.
C C ,, Charging Cylinder.
S ,, Spear.
S H ,, Spear Head.
D C ,, Drawing Cylinder.
S P ,, Swing Plate.
S P R ,, Swing Plate Rods.
H W ,, Hand Wheel.
B ,, Bell.

These several requirements have been very ingeniously worked out and applied by Sir William Arrol and Mr. William Foulis in their patent hydraulic charging machine, shown by the accompanying figure, as follows:—

(1) The whole machine with its load of coal is supported on a pair of rails, placed parallel to the retort bench, and moved by travelling gear T G. In the illustration ordinary hand gear is shown, but in the latest machines a hydraulic motor of special design has been applied with excellent results.

(2) A pipe is fixed the whole length of the retort bench at a convenient height and distance therefrom, and it is always kept charged with water from the accumulator at the full working pressure of 400 lbs. to the square inch. At suitable distances along this pipe there are attached ordinary armoured flexible hose pipes, which may be connected to the pressure swivel P S. From this swivel copper pipes lead to the lifting cock L C and slide valve S V.

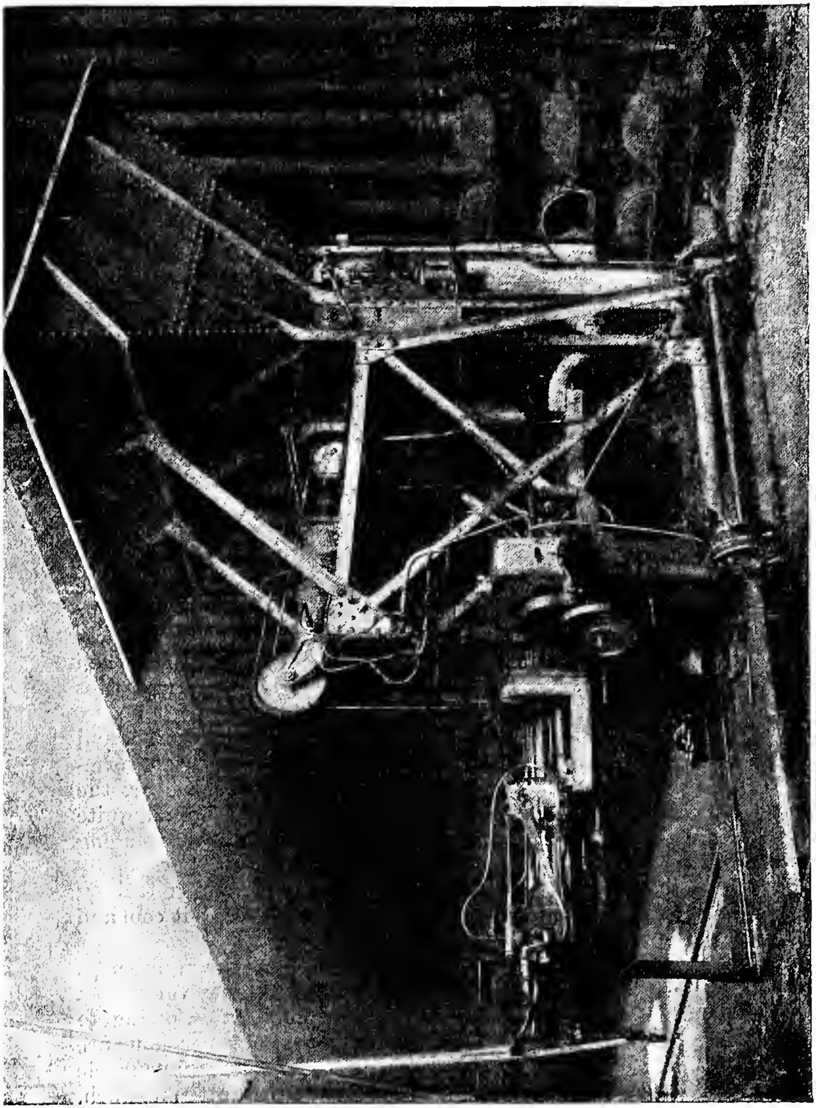
(3) The beam carrying the shoot mouth, shoot rod S R, spear S, charging cylinder C C, and drawing cylinder D C, is raised and lowered by turning the lifting cock L C to the right or left, and thereby admitting water to the beam lifting cylinder B L C.

(4) When the shoot mouth S M is fairly opposite one of the retort mouths it is pushed forward by the shoot-rod lever S R L, and the hand lever H L is turned to admit water to the slide valve S V and work the charging cylinder C C, so as to push the spear S and spear head S H into the retort, having a quantity of coal in front.

(5) The coal hoppers are made to contain from 2 to 5 tons.

(6) The feeding gear consists of an open coal drum C D, which is divided into segmental compartments, each of which can contain a certain quantity of coal. It is turned through one or two divisions at regular intervals by a hydraulic ram with rack and ratchet gear, so as to permit the desired quantity of coal to fall into the shoot. A plate, which acts as a flap valve, is so fixed in front of the drum by a lever and weight, that it presses against the face of the drum, and prevents the small coal from falling down past the face of the drum.

(7) The coal falls from the hopper in front of the pusher plate or spear head S H, and is delivered into the retort by a series of six or seven successive strokes, each stroke being shorter than the previous one. This is accomplished by means of a shaft carrying a set of stops placed on the beam alongside the spear S. This shaft is automatically turned a certain amount during each return stroke, so as to bring the stops into position in rotation. The forward and return strokes of the spear are caused by two



General View of Arrol-Foulis Charging Machine.

hydraulic rams working in the cylinders C C and D C, which are regulated by the hand lever H L. This lever also serves to lower the spear head when entering the retort for pushing in the coal, and to raise it clear from the bottom of the retort during the return stroke. It also causes the revolution of the stopper shaft. The inclined form of the spear head and these successive strokes ensure the charge being evenly distributed along the retort, and the depth is regulated by putting in more or less coal at each stroke, as previously explained.

(8) The platform P is placed at the back of the machine, so that the attendant has the levers within easy reach, and he is placed as far as possible from the heat of the retorts.

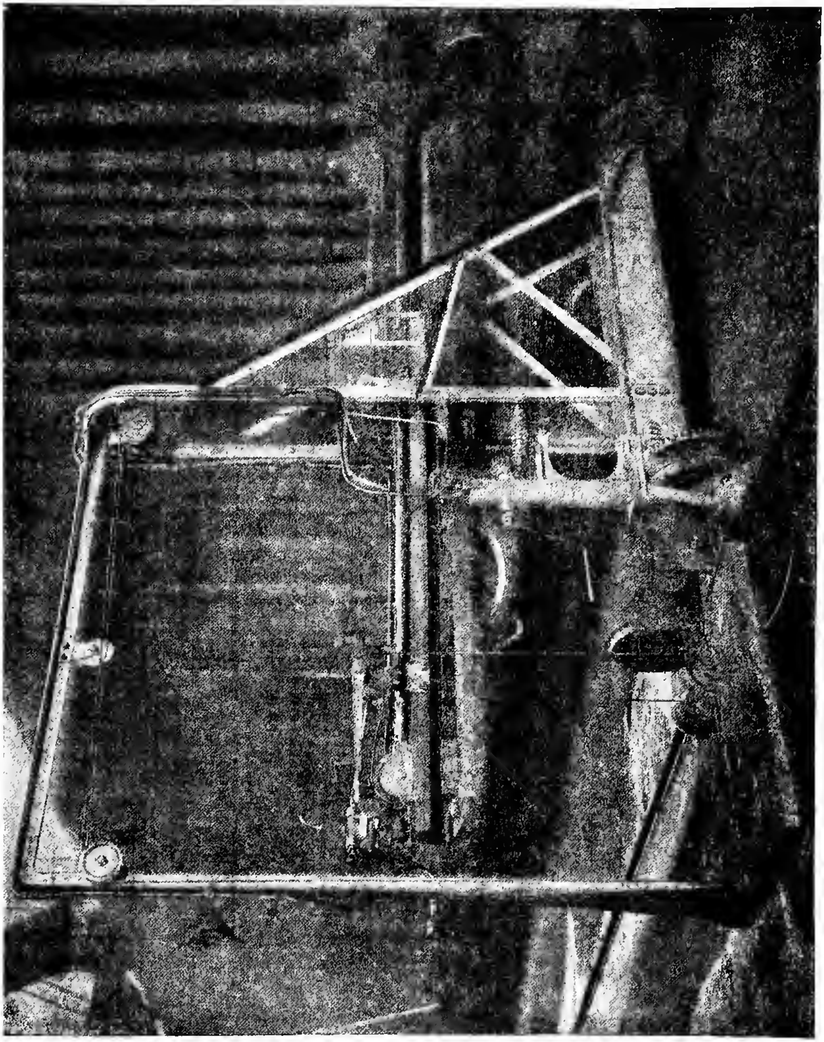
(9) Great attention has been given to the several details of the machine, so as to render it as durable as possible. The reason for adopting this comparatively low pressure is, that the power required for the different motions is so small that if a higher pressure were employed the rams would be inconveniently small. In fact, with 400 lbs. pressure it had been found in some instances that rams of only 1¼ inch diameter gave ample power.

Foulis' Withdrawing Machine for Gas Retorts.—After the coal has been carbonised and converted into coke, it has to be withdrawn from the retorts before putting in a new charge. This is done by a machine which is similar to that used for charging, and travels along the same rails. The withdrawing machine consists of a frame supported on a truck and carrying a rake for pulling out the coke. The head of this rake can swing backwards into a horizontal position so as to clear the coke when being moved inwards, but goes back to the vertical and dips into the coke on its return stroke.

The beam B carrying the rake can be raised or lowered, and the rake moved out and in, in the same manner as the corresponding parts of the charging machine. As, however, the withdrawing machine is comparatively light, it is found quite sufficient to use hand travelling gear T G to make it move along the rails.

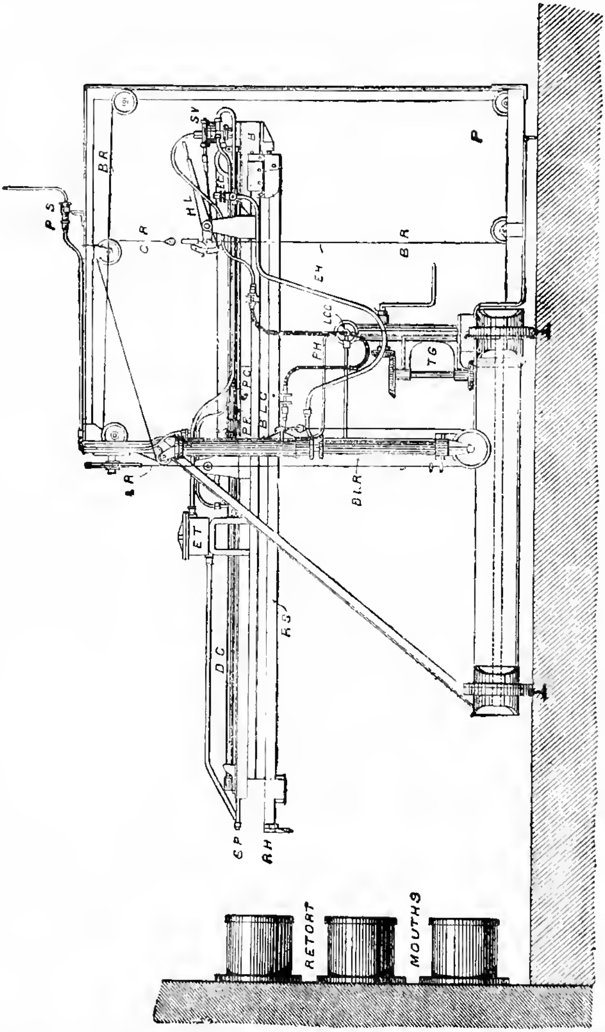
The exhaust water is led to a tank E T, from which it flows to a spray pipe S P and plays on the rake head to keep it cool and help to quench the coke.

Results of Working.—Mr. Biggart says that the number of retorts charged or drawn per hour by these machines varies to a considerable extent in actual work. In some cases, owing to special circumstances, not more than twenty-four per hour are available for each machine; while in other more favourable instances as many as forty-eight per hour are allotted to each, and even with this larger number a reasonable time remains for



FOULIS' WITHDRAWING MACHINE FOR GAS RETORTS.

rest for the attendants at the end of each hour. The labour of charging the retorts and withdrawing the coke is much lightened by these mechanical means, and the number of retorts charged and drawn for each man employed is largely increased. It



Foulis' Withdrawing Machine for Gas Retorts.

might at first be imagined that coal placed in retorts in only six to eight large charges by the machine would not be so evenly laid as a much larger number of smaller charges put in by hand. The machine, however, lays the coal by far the most evenly, owing partly to the shape of the pusher head, which is bevelled so as to allow the small ridge of coal raised in pushing forwards to fall back when its support is removed on the withdrawal of the pusher head. Another advantage possessed by machine work over hand labour is that the charging is done more quickly, and thus there is a diminished loss of gas before the retort doors are closed.

Apart from any other consideration, the mechanical charger could not fail to prove beneficial in view of the greatly improved conditons under which it enables work of a most trying nature to be carried on. The old method of hand charging was a severe ordeal for the stokers, requiring great exertion to get through the work in the shortest time possible, while exposed throughout to a high heat. Such adverse conditions are now entirely done away with where mechanical stoking obtains. The single lever by which the whole of the operations are controlled is worked from such a position that the attendant is quite removed from the discomfort of close proximity to a high heat, while at the same time the former severe bodily exertion is replaced by light and easy work. Even greater improvements in the conditions of labour arise from the introduction of the drawer, which accomplishes, under all the better conditions attending the use of the charger, work of a still more trying nature. The withdrawing of the hot coke from the retorts was work for which even the stokers themselves, accustomed as they were to it, admitted that mechanical appliances were required. Here again all is worked by a single lever, in such a position as to remove the attendant from the former discomforts of withdrawing the coke at a white heat at the mouth of the open retort.

INDEX TO PARTS.

T G	for	Travelling Gear.	P H	for	Pressure Hose.
P S	,,	Pressure Swivel.	P C	,,	Pushing Cylinder.
L C C	,,	Lifting Cylinder Cock.	P R	,,	Pushing Ram.
B	,,	Beam.	R S	,,	Rake Spear.
B L C	,,	,, Lifting Cylinder.	R H	,,	,, Head.
B L R	,,	,, ,, Ram.	E C	,,	Exhaust Cock.
L R	,,	Lifting Rope.	E H	,,	,, Hose.
B R	,,	Balancing Ropes.	E T	,,	,, Tank.
C R	,,	Check Rope.	S P	,,	,, Spray Pipes.
H L	,,	Hand Lever.	D C	,,	Drawing Cylinder.
S V	,,	Slide Valve.	P	,,	Platform (for Attendant).

Lecture III.—Questions.

1. Sketch and describe the general arrangement of pumping engines and accumulator as used for supplying hydraulic power to the mechanical stoking appliances in a gas works.

2. Make a vertical section of an accumulator, and explain the manner in which this apparatus enables us to store up and give out energy. If the ram of the accumulator be 17 inches in diameter, what should be the load in order to obtain a water pressure of 700 lbs. on the square inch? *Ans.* 70·9 tons.

3. Sketch and describe some arrangement of an hydraulic accumulator by which a pressure of 10 tons to the square inch can be obtained for testing purposes, with pumps working at a pressure not exceeding 3 tons to the square inch.

4. Describe, without going into detail, the engines, pumps, accumulator, and one or two of the appliances likely to be used by customers of an hydraulic company.

5. If you desire to obtain great pressure with a small dead load, what form of accumulator would you employ? Give a vertical section and a description of the construction and action of the accumulator you have selected. Why is it inadvisable to use one with a very small ram?

6. If you desire to obtain a pressure of 1,200 lbs. per square inch, and you are limited to a dead weight of 1 ton, what effective area would you require, and what would be the diameter of the larger part of the ram if the smaller be 6 inches in diameter, in the case of a differential accumulator having a total efficiency of 90 per cent.? What would be the diameter of the ram if you used a solid one?

7. Sketch and describe Brown's Steam Accumulator. Mention any advantages and disadvantages which you consider it possesses with respect to other forms of accumulator.

8. Suppose the effective steam pressure in the steam cylinder of a Brown's Accumulator to be 60 lbs. on the square inch, and that you require a water pressure of 1,000 lbs. per square inch, with a ram of 20 square inches in cross section, what will be the diameters of the ram and steam piston if 2 per cent. be lost in friction?

9. Sketch the construction and describe the action of the Arrol-Foulis hydraulic apparatus for charging gas retorts. Mention the several advantages of employing this machine as against hand labour.

10. Sketch the construction and describe the action of the Foulis hydraulic apparatus for withdrawing the coke from gas retorts. Mention the several advantages of employing this machine as against hand labour.

11. What is the use of an intensifier or intensifying accumulator in the working of hydraulic machinery? Sketch such an apparatus, and explain fully its principle and construction: give also one example of the application of the intensifier to hydraulic machinery.

12. A differential accumulator, loaded to 1,400 lbs. per square inch, has a 5-inch spindle and a ½-inch bush, and supplies a riveter ram of 9 inches diameter through 30 feet of 1-inch piping. Find the "equivalent mass" at the riveter ram on the assumption that no water is pumped into the accumulator during the working stroke of the riveter ram.
(C. & G., 1902, H., Sec. C.)

13. Sketch an automatic governor suitable for use on a main pumping engine which, by operating on the steam valve, reduces the speed of the pump when the pressure in the delivery pipe exceeds a certain amount.
(C. & G., 1902, H., Sec. C.)

N.B.—*See Appendices B and C for other questions and answers.*

LECTURE III.—A.M.INST.C.E. EXAM. QUESTIONS.

1. Give a short account of the arrangements for distributing power by pressure water. Sketch an accumulator. Find the energy stored in foot-pounds per cubic foot of water, in an accumulator loaded to 700 lbs per square inch. (I.C.E., *Feb.*, 1898.)

2. Sketch, dimension, and describe a proper packing gland for an accumulator 12 inches in diameter and 4,000 lbs. per square inch water pressure. (I.C.E., *Feb.*, 1901.)

3. What amount of energy is stored in p lbs. of water at q lbs. per square inch pressure? A differential hydraulic press has two stuffing boxes in line and a vertical rod passes through both, the diameters of the pistons at its lower and upper ends being 5 inches and 8 inches respectively. What load does the press take when supplied with water at a pressure of 1,400 lbs. per square inch? What store of energy can it take in, in foot-pounds, if it can rise through 10 feet, and how many gallons of water are required for the operation? (1 gallon of water = 10 lbs.) (I.C.E., *Feb.*, 1901.)

4. Sketch any form of direct-acting compensated steam pump, explaining the action of the compensator by reference to combined diagrams. (I.C.E., *Oct.*, 1902.) (*See Brown's Steam Accumulator*).

N.B.—*See Appendices B and C for other questions and answers.*

LECTURE IV.

HYDROKINETICS.

CONTENTS.—Energy of Flowing Water—Bernouilli's Theorem—Jet Pumps, Injectors and Ejectors—Hydraulic Ram—Example I.—Velocity of Efflux and Flow of Water from a Tank—Measurement of a Flowing Stream—Rectangular Gauge Notch—Thomson's Triangular Notch—Measurement of Head—Measurement of Large Streams—Horse-Power of a Stream—Vortex Motion—Free Vortex—Forced Vortex—Pressure due to Centrifugal Force—Reaction of a Jet—Reduction of Pressure Round an Orifice—Impact—Loss of Energy—Resistance of a Pipe—Hydraulic Mean Depth—Loss of Head due to Friction in Water Pipes—Examples II. and III.—Notes on Measurement of Streams, &c.—Questions.

Energy of Flowing Water—Bernouilli's Theorem.—When a liquid is flowing in a pipe or channel, it possesses kinetic energy in virtue of its motion in addition to the potential energy due to its position and pressure; and the total energy is the sum of these three.

Let v = Velocity of the liquid.
,, h_1 = Length of its pressure column.
,, h_2 = Height above the datum level.
,, H = Height of the free surface of the still water above the datum level.
,, m = Mass of the portion of the liquid under consideration.
,, g = Acceleration due to gravity.
,, f = Frictional loss of head.

Then,

The energy of pressure $= m\,g\,h_1$
,, of position $= m\,g\,h_2$
And, ,, of motion $= \frac{1}{2}\,m\,v^2.$

Hence, The total energy $$= m\,g\left(h_1 + h_2 + \frac{v^2}{2\,g}\right). \quad \text{(I)}$$

Or, Energy per unit mass $$= g\left(h_1 + h_2 + \frac{v^2}{2\,g}\right) \quad . \ (\text{I}_a)$$

It follows from the principle of the *Conservation of Energy* that so long as no work is spent on friction, this total remains constant whilst the water flows along the pipe or channel. Therefore, for a frictionless liquid in which there are no eddies :—

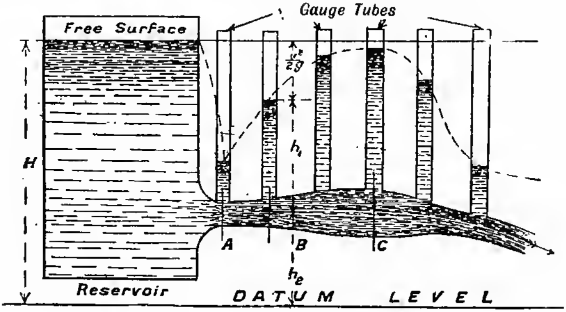
$$h_1 + h_2 + \frac{v^2}{2\,g} = \text{a constant} = \text{H}. \quad . \quad . \quad . \quad \text{(II)}$$

If, p = Pressure per unit area at any point in the liquid,
And, w = Weight of unit volume of the liquid,

Then, $h_1 = \frac{p}{w}$;

$$\text{Hence, } \frac{p}{w} + h_2 + \frac{v^2}{2g} = \text{a constant} = \text{H.} \qquad \text{(III)}$$

This equation is known as *Bernouilli's Theorem.*



PRESSURES IN A FRICTIONLESS PIPE.

If a certain amount of energy mgf be absorbed by friction between some vertical datum section such as A and the section under consideration which may be at B or C.

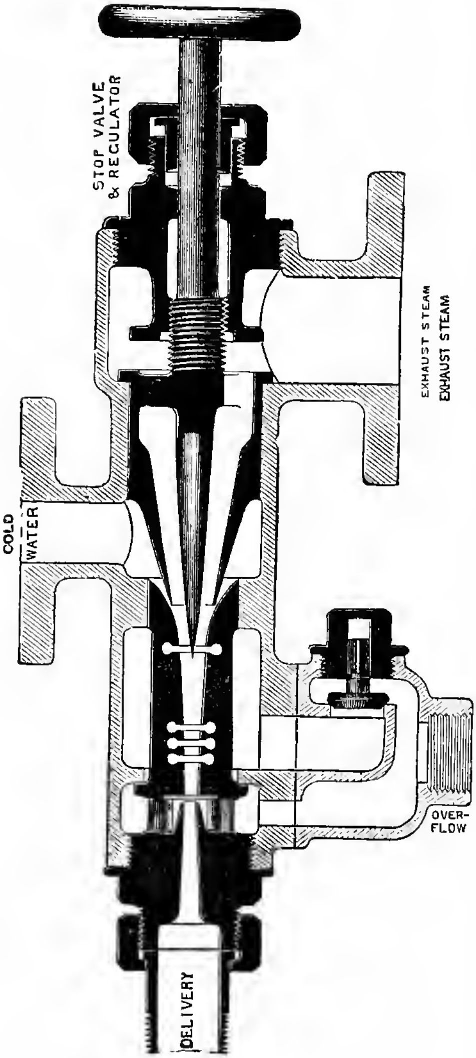
$$\text{Then, } \frac{p}{w} + h_2 + \frac{v^2}{2g} + f = \text{a constant} = \text{H.} \qquad \text{(IV)}$$

Let a = The cross area of the pipe at any section.
,, v = ,, velocity of the water at the same section.
,, Q = ,, quantity or volume of water passing every section in unit time. This volume is constant for all sections during a steady flow of the liquid.

$$\text{Then, } a\,v = Q\,;\ \text{or, } v = \frac{Q}{a}. \qquad \text{(V)}$$

This shows, that the velocity is great when the cross section of the pipe is small, and *vice versâ.*

In the figure, we have shown small vertical gauge tubes placed at intervals A, B, C, &c., along the pipe, in order to indicate the pressure of water at these points by the height to which it rises in them. It will be observed that where the pipe is level or nearly so, the pressure is greatest where its cross section is largest and consequently the velocity is least. This follows directly from



Exhaust Steam Ejector.

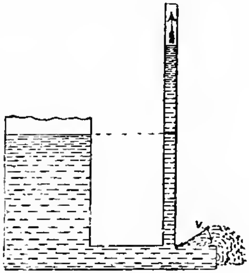
Bernouilli's theorem, since the height h_2 above the datum level is practically constant.

Jet Pumps, Injectors, and Ejectors. — From the previous illustration and explanation of the alteration in the pressure of water as it flows through a pipe of varying size, it will be readily understood that if a stream of water be rapidly forced through a tapered passage, its pressure may be so lowered below the surrounding atmosphere, that it can draw in more water from another source connected to the smaller end of the taper. If the whole of the water be then conducted along a gradually enlarged passage, its pressure will increase and the outflow can take place at a higher level than the intake of the induced stream, but lower than the free surface of the driving water. The late

Prof. James Thomson, of Glasgow University, designed his jet pump upon this principle, and Prof. Bunsen used a jet of water to produce a vacuum in his air pump. Steam jets, compressed air, and water under pressure, have frequently been used to create a blast of air, to feed petroleum into furnaces, to produce a sand blast for engraving or cleaning purposes, and to transfer granular materials from one position to another.

Injectors for feeding steam boilers with water also work upon this principle; but since they are greatly assisted by the condensation of the steam as it comes into intimate contact with the suction water, the latter can be forced into the same boiler or another vessel having the same pressure as, or even a higher pressure than, that which supplies the injector with steam.* Exhaust steam ejectors also depend upon the above action, and are sometimes used to replace both the ordinary jet condenser and air pump in connection with condensing engines. As will be seen from the accompanying figure, when the regulating stop valve is screwed upwards by turning the hand-wheel, the exhaust steam from the engine cylinder enters by the right-hand upper pipe and mixes with cold water coming through the left-hand one, thereby becoming condensed and producing the desired vacuum. The combined condensed steam and condensing water then flow from the delivery pipe into the hot well, whilst any throttled discharge, or some of the live steam that may have been used for blowing through and starting the ejector, can escape into the same place by the overflow pipe. As we shall prove further on, apparatus of this kind cannot have a very high efficiency since the mixing up of quick and slow moving streams results in a considerable loss of mechanical energy.

Hydraulic Ram.—This apparatus was invented about 100 years ago by a Frenchman named Montgolfier. It is one of the simplest, most durable, and efficient machines for raising water to a greater height than the source of supply. The energy stored up in water descending from a comparatively low elevation is utilised to raise part of the same water to a much higher level, of from three to thirty times the vertical height of the original fall. The principle upon which the apparatus works will be understood from



PRINCIPLE OF THE HYDRAULIC RAM.

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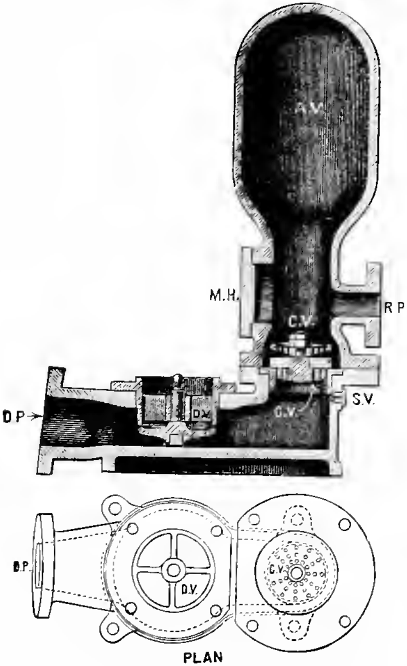
[*] For a description of Giffard's and other steam injectors see the author's *Text-Book on Steam and Steam Engines.*

a consideration of the foregoing figure. If the valve V be held down firmly on its seat, and the left-hand vessel be filled with water to a certain height, it will rise to the same level in the right-hand open pipe. If the valve be now released for a short time, water will flow under the action of gravity along the horizontal passage and escape at the open valve with a velocity proportional to the square root of the "head" or vertical height of the free surface above the valve. On suddenly closing the valve, the kinetic energy of the moving water will be partly spent in raising the right-hand column to a greater height than the free surface of the water in the left-hand vessel. Now, if we introduce a check valve at the foot of the long column, so as to prevent this water from falling down again, and an air vessel to act as a cushion, we can repeat the operation continually, so as to produce a flow of water up the pipe.



Hydraulic Ram, by the Glenfield Company, Kilmarnock.

The machine, as made and supplied by the Glenfield Company of Kilmarnock, is illustrated by a vertical section and plan in the accompanying figure. Water flows from a cistern, tank, pond, or dam, through a cast- or wrought-iron pipe, technically called the drive pipe D P, to a hollow casting containing two valves. The first of these is named the escape or dash valve D V, which opens downwards, and the other the check valve C V, which opens upwards. Over the latter is fixed an air vessel A V, having a manhole door M H to the left and a delivery pipe, which is technically termed a rising pipe R P, to the right. If the apparatus and all the pipes are duly connected to the supply and delivery tanks, and the dash valve D V be held up, until the water from the source of supply has filled not only the drive pipe D P, but also risen through the check valve C V and rising pipe R P to nearly the same level as the free surface in the supply tank, the whole will remain motionless or in a static condition. If we now depress the dash valve D V, and then let it go, the machine will immediately begin to work, and continue to work automatically without any attention or even oiling for years, until stopped by some accident or by the wearing out of one or both of the valves. Of course, the supply of water must be maintained, so that the drive pipe is always kept full. This pipe should not be throttled in any part, and the weight* on the dash valve must be so carefully adjusted, that it will just overcome the internal pressure—*i.e.*, drop from its seat—and permit water to escape thereat. Then, the acceleration produced by gravity on the water coming down the drive pipe very soon produces a greater pressure on the dash valve than that due to the mere static pressure. This increased force suddenly raises it again to its seat, when the kinetic energy which has been imparted to the water lifts the check valve and forces some water into the air vessel. Whenever this kinetic energy has been spent,* the compressed air in the air vessel, together with the weight of the check valve, causes it to close, and immediately thereafter the dash valve automatically opens. The same cycle of operations takes place over and over again, the air in the air vessel gets more and more compressed, and water rises higher and higher in the rising or delivery pipe, until it issues as a continuous stream from its mouth into the cistern or receiving tank. From this tank it may be drawn off at pleasure for all the various uses of a mansion-house or farm steading, &c.

The air vessel plays two important parts in each cycle of the operations of this interesting and useful apparatus. (1) The air contained therein acts as a cushion by minimising the water hammer action, which would otherwise stress the various parts, and tend to break the joints. (2) The air acts as a store of energy by taking up, during its compression, a part of the kinetic energy of the water, and then giving out the same gently, thus producing a constant flow of water through the delivery pipe. If the vertical height of the column of water in the rising pipe be about 34 feet above the check valve, the pressure per square inch on the upper surface of this valve will be one atmosphere, or, say, 15 lbs. on the square inch, and the air in the air vessel will be compressed to nearly that pressure,

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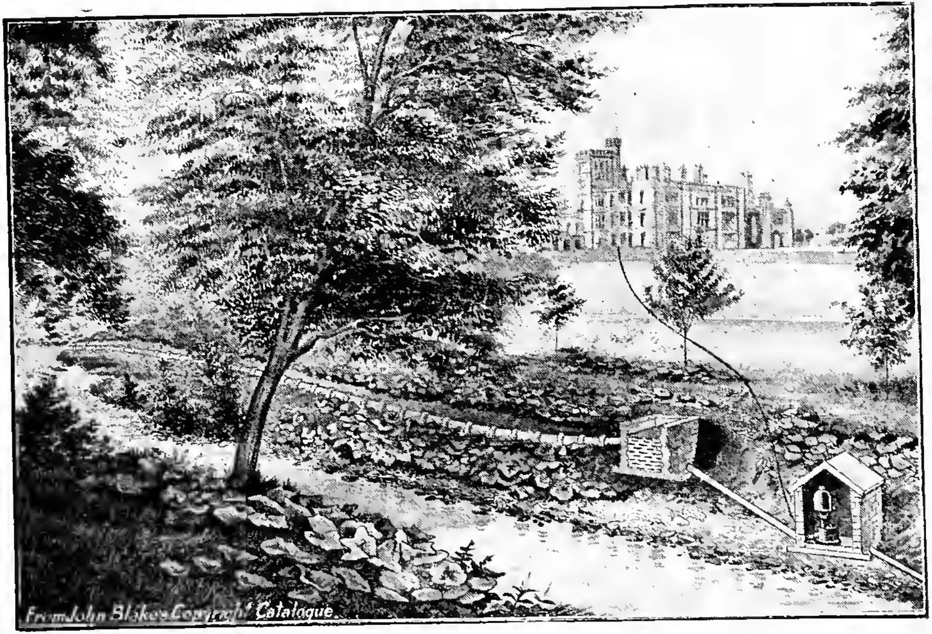
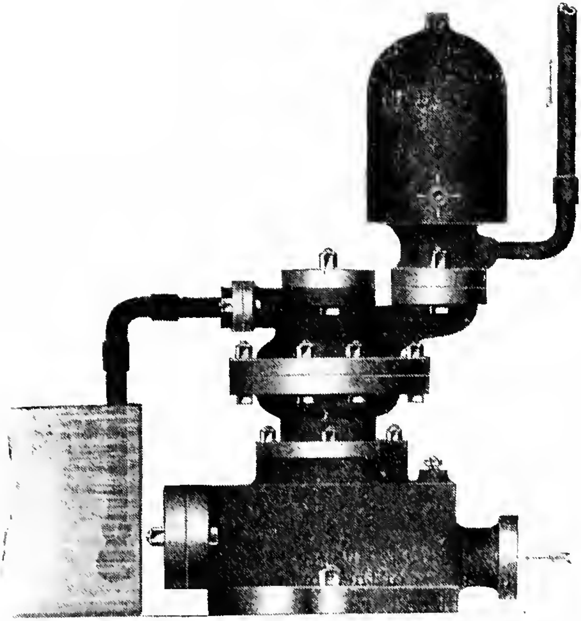
* See Note at end of this Lecture.



From John Blake's Copyright Catalogue.

BLAKE HYDRAM RAISING WATER FROM A RIVER WITH A FALL OF 4 FEET TO A VERTICAL HEIGHT OF 100 FEET.



Blake "B" Ram for Working with Impure Water whilst raising Pure Water from a Spring or Shallow Well.

and therefore occupy about half its original volume. If the column be 68 feet high in the delivery pipe, the pressure on the valve will be about 30 lbs. on the square inch, and the air in the air vessel will occupy one-third of its original volume, and so on. Hence, it is necessary to proportion the size of the air vessel to the vertical height through which the water has to be forced in the delivery pipe. Besides this, air becomes absorbed by water, and in a short time the air vessel, if small, would become entirely filled with water. The air vessel may, however, be kept charged with air in a very simple manner by the introduction of a snifter valve S V, screwed into the ram casing, immediately below the check valve. In its simplest, and probably its most efficient form, it consists of a brass plug with a very small hole drilled through its axis. Every time that water is forced through the check valve a very small quantity also passes through this tiny opening in the snifter valve; and each time that the check valve is forced down upon its seat a rebound or reaction of the water takes place, and produces a partial vacuum immediately underneath the check valve. Consequently, a little air is forced into this vacuum by the atmospheric pressure, and this air is carried up into the air vessel at the next stroke or pulsation, thus keeping up the necessary supply for effecting a continuous flow of water into the receiving tank. If everything about this machine is thoroughly tight and in good working order, and the valves are made of the best proportions and weights, an efficiency of from 80 to 90 per cent. can be obtained therefrom, and it has been found possible to work it with a minimum driving head of only three feet. The several causes for loss of efficiency are:—

(1) Eddies caused by the sudden stoppage of the water's motion.

(2) The friction of the water passing along the drive pipe D P, and the casing of the apparatus.

(3) The weight and friction of the dash valve D V, which has to be lifted at each stroke or pulsation.

(4) The weight and friction of the check valve C V, which has also to be lifted at each stroke.

(5) The slip of the check valve C V—*i.e.*, a slight quantity of water may slip back past this valve when in the act of closing.

(6) The friction of the water passing along the rising pipe R P.

(7) Any defects of tightness in the faces of the dash and check valves.

The sudden closing of the dash valve is only necessary to prevent the water spending a large portion of its energy in friction at the restricted orifice while it is closing. Large rams are now made with special valves to stop the water gradually without throttling it, and so avoid the shocks caused by sudden closing without losing anything in efficiency.*

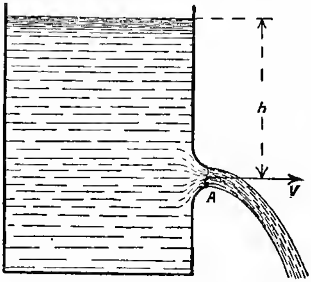
Example I.—If 1,000 lbs. of water pass per minute through the drive pipe under a head of 6 feet, and 60 lbs. of water are delivered into the receiving tank, which is 87 feet above the check valve, what is the efficiency?

$$\text{Efficiency} = \frac{\text{work got out}}{\text{work put in}} \text{ (in the same time).}$$

$$\therefore \qquad \text{Efficiency} = \frac{87 \times 60}{6 \times 1000} = \cdot 87$$

Or, $\qquad$ **Efficiency = 87 per cent.**

If the length of the drive and rising pipes be considerable, and if there be many bends and much throttling of the passages, then the efficiency will thereby be reduced to a considerable extent. By a simple modification of the ram shown in the illustration, river or impure water may be made to raise spring or pure water; the two waters are separated by a diaphragm, and the pumping action actuates two valves, the one being a suction and the other a delivery valve.



Efflux of Water from a Circular Orifice.

Velocity of Efflux and Flow of Water from a Tank.—Consider a jet of water issuing from a small circular orifice A at a depth h below the free surface of the water in the tank. If we take the datum line at the orifice, then inside the vessel, where the water is still, the energy is entirely potential and equal to $m\,g\,h$; whereas, it is all kinetic just outside the opening and amounts to $\frac{1}{2}\,m\,v^2$.

Hence, $\qquad \frac{1}{2}\,m\,v^2 = m\,g\,h$; or, $v^2 = 2\,g\,h$;

i.e., $$v = \sqrt{2\,g\,h}. \qquad \text{(VI)}$$

It will be observed from this equation (VI) that the velocity

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* See § 30 of Gordon Blaine's *Hydraulic Machinery* for an illustrated description of Mr. H. D. Pearsall's improved hydraulic ram.

v is the same as that attained by a body in falling freely from a height h; and further, that it would be the same even if the water had no free surface, so long as the pressure at the level of the orifice was equal to that due to a head h.

If there be a loss of head f due to friction and eddies formed by the water in passing through the orifice,

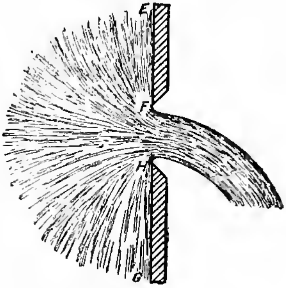
$$\text{Then,}\quad \tfrac{1}{2} m v^2 = m g (h - f)\text{; or, } v = \sqrt{2 g (h - f)}. \qquad \text{(VII)}$$

If, a = The cross area of the orifice in square feet.
r = „ radius of the orifice in feet.
v = „ average velocity of the water in feet per second.
h = „ head of the water in feet.
g = „ acceleration due to gravity or 32·2 feet per second per second.
And Q = „ quantity of water flowing out from the orifice in cubic feet per second.

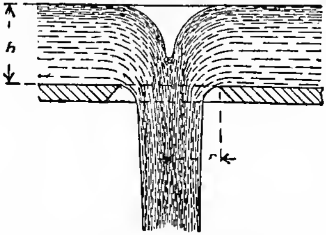
We find from equations (V) and (VI) that :—

$$Q = a v = a \sqrt{2 g h} \text{ cubic feet per second.}$$

This formula is very nearly true for a small tapered opening, but with a flat one, as shown in the next figure, the cross area of the stream where the stream lines are parallel, at a short distance outside the opening, is less than that of the orifice. The ratio of these two areas is called the *coefficient of contraction*, and it is found experimentally for a flat opening to be 0·64. The contraction is caused by the water flowing along the inner flat surfaces E F and G H and then leaving them at a tangent. It has also been found by experiment that for a sharp-edged orifice the velocity v is only



Efflux of Water from a Vertical Flat Orifice.



Efflux of Water from a Horizontal Flat Orifice.*

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* By mistake the figure has been drawn with a vortex, but when measuring water we must prevent the formation of a vortex by putting in radial blades.

$0 \cdot 97 \sqrt{2 g h}$. Hence, the actual flow for *a small circular orifice* will be :—

$$Q = 0 \cdot 64\, a \times 0 \cdot 97 \sqrt{2 g h} = 0 \cdot 62\, a \sqrt{2 g h}$$

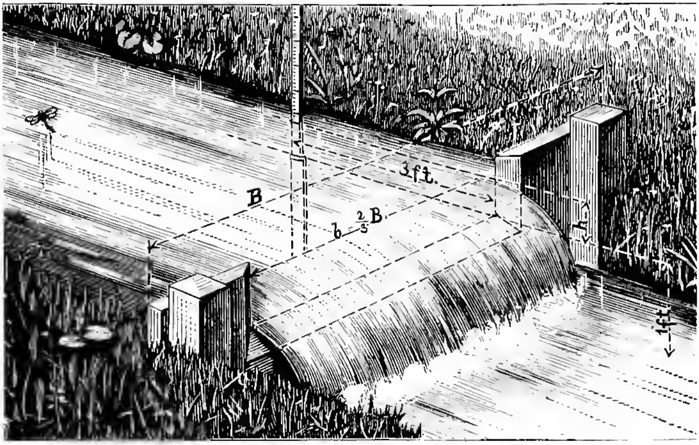
$$\therefore \qquad Q = 0 \cdot 62\, \pi r^2 \sqrt{2 g h} \text{ cubic feet per second.} \qquad \text{(VIII)}$$

In measuring a small flow of water by this method, it is run into a tank having a carefully made clean cut orifice of known area, until the surface level is just sufficiently above the outflow to cause the water to run out as fast as it runs into the tank. This difference of level, or head h, is measured and the above formula applied.

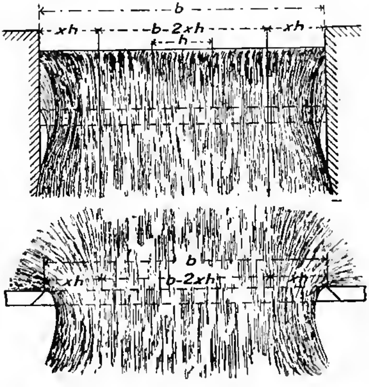
Accurate results cannot be obtained from a large circular opening placed in the side of a tank, because the parts of it would have different depths from the free surface, and consequently the water would have different velocities at these parts. If we, however, put the orifice in the bottom of the tank, as shown in the right-hand figure, then the velocity will be approximately the same at all parts of the opening, and we can enlarge it so as to measure a much greater flow of water.

Measurement of a Flowing Stream.—In order to ascertain the available power from a stream or river, as well as to test the efficiency of a hydraulic installation, it is of the first importance to determine the rate of flow—*i.e.*, the number of cubic feet or gallons of water passing a given point per unit of time. At first sight, this would appear to be a very simple matter; but, as will be shown, it is not so easy to do so with accuracy, for several special precautions have to be observed and constants obtained.

Rectangular Gauge Notch.—When we have to measure large quantities of moving water, then such orifices as we have previously dealt with are quite unsuitable. In such cases, it is usual to pass the whole of the water over a special form of weir or gauge notch. This consists of a board placed across the stream between stakes and carefully puddled, so that all the water must flow over it. The top bevelled edge of this board must be well above the surface of the water on the delivery side. The length of the notch is usually less than the width of the stream, and the board should be continued from each end in a straight line (as shown in the perspective view) so as to give definite conditions; or parallel guides may be placed in the stream before the gauge board. In the former case, the inflow of water at the ends of the notch board causes end contraction, while in the latter, this effect is avoided. In each case, contraction of the stream is caused by the water flowing upwards from E to F, as shown by the sectional figure a few pages further on. Further, the surface directly over the notch board is lower than that of the still water in the pond above the board. If, however, we neglect these effects we see that with different lengths of notch board the total flow will be proportional to its length. Now, consider any horizontal strip of the cross section of the stream over the notch (say, one-thousandth of its total depth); then, if we increase the depth of the stream over the notch, the vertical width of this strip and consequently its area will be proportionally increased, as well as its depth below the free surface of the water. But the velocity of the water passing through this strip varies as the square root of its depth, and the quantity as the area multiplied by the velocity. This result will hold good for each elementary strip, and will, therefore, apply to the whole stream.



Perspective View of a Rectangular Gauge Notch.



Front View and Plan of a Rectangular Gauge Notch.

Hence, $Q \propto a\,v \propto h \times \sqrt{h} \propto h^{\frac{3}{2}}$ for different depths;

And, $Q \propto b$ for different breadths of stream at the notch;

$\therefore$ $Q = k\,b\,h^{\frac{3}{2}}$ for a rectangular notch. . . . . . . (IX)

Here, k is a *coefficient of discharge* which must be found by experiment. This equation, however, would not give us accurate results if the proportions of the stream passing the notch were much different from that used to determine the constant k. With a very long shallow notch a considerable error will arise from the fact, that the water may adhere to the horizontal bevelled edge, and with a very deep narrow notch a similar effect would be produced by the bevelled sides. The above formula is often used and is quite correct for similar streams; but if the flow is variable, the depth will change with the flow of the water while the breadth remains constant, so that the proportions of the stream obtained with different flows in a gauge notch of this kind are not the same. (*See Note 1 at the end of this Lecture for Mr. Ritchie's Rule.*)

The late Professor James Thomson showed how we may obtain a formula which will apply to all ordinary proportions of rectangular notches. In the central part of the stream the lines of flow are practically parallel and unaffected by the sides of the notch. Consequently, the water passing through this part will be proportional to its breadth. Suppose the influence of each end to extend perceptibly to a distance $x\,h$ from it. Then the breadth of the central part will be $b - 2\,x\,h$. Consider a portion of this central part whose breadth is equal to h. This will be a square and therefore similar for different sizes of stream. Hence, since the area is proportional to h^2 and the velocity to $\sqrt{h}$, the flow through this square will be :—

$$k_1 \times h^2 \sqrt{h} \text{ or } k_1\,h^{\frac{5}{2}}$$

where k_1 is a constant.

Consequently, the flow through the whole of the central part $(b - 2\,x\,h)$ will be :—

$$\frac{b - 2\,x\,h}{h}\,k_1\,h^{\frac{5}{2}} = k_1\,(b - 2\,x\,h)\,h^{\frac{3}{2}}.$$

If we now imagine the two side portions to be placed together, we will get another stream which will be of similar form whatever its actual size may be; for, it will always be a rectangle of depth h and length $2\,x\,h$. Consequently the flow through this will be:—

$$k_2 \times 2\,x\,h \times h^{\frac{3}{2}} = 2\,k_2\,x\,h^{\frac{5}{2}}$$

where k_2 is another constant.

Hence, the flow of the whole stream will be :—

$$Q = k_1 (b - 2xh) h^{\frac{3}{2}} + 2 k_2 x h^{\frac{5}{2}}$$

$$\therefore \qquad Q = k_1 \left\{ b - \frac{2x(k_1 - k_2)}{k_1} h \right\} h^{\frac{3}{2}}.$$

If we write c for $\frac{x(k_1 - k_2)}{k_1}$ which is a constant, we get:—

$$Q = k_1 \{b - 2ch\} h^{\frac{3}{2}}. \quad . \quad . \quad . \quad . \quad . \quad (X)$$

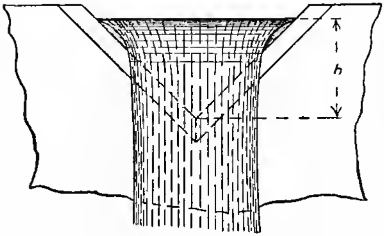
If one of the sides of the stream had a guide board we should have had x throughout instead of $2x$, and therefore ch in our final results instead of $2ch$. If both sides of the stream be guided, this term would disappear and we would get the former result.

Mr. Francis, an American engineer, deduced the following empirical formula from a large number of experiments which he made :—

$$Q = 3{\cdot}33 \left(b - \frac{1}{10} n h\right) h^{\frac{3}{2}} \text{ cubic feet per second.} \qquad (X_a)$$

Here n is the number of end contractions (viz., 2, 1, or 0, as explained above), and the units employed are feet and seconds. It should be noted that this equation is of the same form as the previous one, and that neither is applicable to a notch whose length is less than $2xh$.

Thomson's Triangular Notch.—Professor James Thomson proposed and used a gauge notch in the form of a right-angled isosceles triangle with its sides equally inclined to the vertical. It has the advantage of giving a similar form of stream whatever may be the size of the notch or the height of the water passing through it, and is, therefore, more accurate for measuring variable streams. As, however, less water is passed for a given height than with the rectangular notch, it is not so convenient for large flows; but by cutting a number of such notches, side by side like the teeth of a saw, considerable quantities of water may be dealt with.



THOMSON'S TRIANGULAR GAUGE NOTCH.

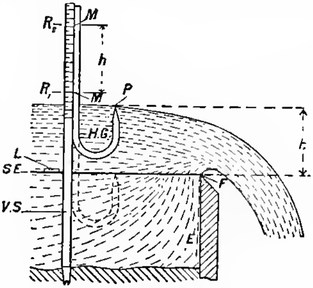
If we consider corresponding elements of two such notches, we see that their areas are proportional to the square of their depths, whilst, as before, the velocities of the water are as the square roots of the depths. Hence, the flow through a **V** notch will be :—

$$Q = k \times h^2 \times \sqrt{h} = k\,h^{\frac{5}{2}} \text{ cubic feet per second.} \qquad \text{(XI)}$$

In this case, the *coefficient of discharge k*, has been found by careful experiment to be 2·64.

Measurement of Head.—When using either of the previously mentioned notches for determining the flow of a stream or river we must ascertain the head *h*, with great accuracy. This may be done by aid of a level, straight edge, graduated staff, and a bent wire or hook-gauge in the following manner :—

Drive the vertical stake V S, into the bed of the stream at a position above the notch where the surface has no appreciable velocity. To obtain such a position the pond above the weir should not be too small. Level a straight edge S E, by the level L, with its lower edge resting on the inner edge F, of the bevelled board and on the point of the hook-gauge. Note the position R_1 on V S, opposite a mark M, on the longer limb H G. This gives us once for all the zero from which to reckon *h*.



Apparatus for Measuring Head of Water.

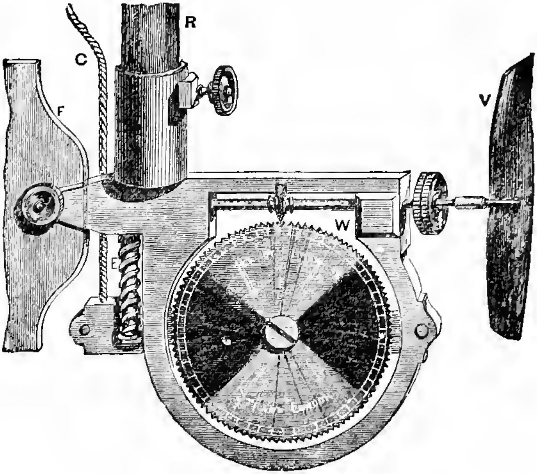
When the water is flowing in the normal condition for making the test, raise H G until the point P just touches the surface of the water and note the reading R_2 on V S opposite M, as shown by the full outlined hook in the figure. The difference between this reading and the former one gives the head *h*.

As may be seen from the previous formulæ any mistake made in determining *h* will produce a larger percentage error in the result with the **V** and rectangular notches than with an orifice in the bottom of a tank. The latter is, therefore, preferred where great accuracy is desired and the quantity of water is not too large, such as when measuring the circulating water used by a steam engine.

Measurement of Large Streams.—When it is inconvenient or impracticable to place a weir gauge in a river, then the flow may be estimated by measuring the cross section *a*, of the river and finding its mean velocity *v*, at that section :—

Then, as before, $Q = a\,v.$

To do this, a number of equidistant points are marked along a rope which is then stretched across the river at right angles to the direction of the current. By means of a graduated pole the depth at each of these points is ascertained from a boat. The results are then plotted to scale and give us a cross section of the river and an estimate of its sectional area. The surface velocity at midstream may be roughly found by noting the time taken for a float to move a given distance down stream, and the *mean* velocity may be taken as 0·65 of this. It is, however, much more accurate to ascertain the velocity at a number of points of the section by means of a current meter and then calculate the mean value. The following illustration shows a current meter made for this purpose by Messrs. Elliott Brothers, London. It has a screw-shaped vane V, which is rotated by the water flowing past it. The revolutions of this vane are counted by the wheel W, which is driven by a worm on the same spindle as the vane. When the apparatus is immersed by means of the rod R to the required depth, with the vane pointing up stream, the cord C is pulled up and kept tight for a definite interval of time. This cord is attached to the end of a lever which carries the bearings of the counting wheel and is pushed down by a spring E. The wheel only gears with the worm when the cord is pulled, and the reading gives the number of revolutions of the vane from which the velo-



ELLIOTT'S CURRENT METER.

city of the water may be deduced. F is a rudder to keep the apparatus pointing directly upstream.

Horse-Power of a Stream.—After having obtained the quantity of water flowing in a stream, we have only to measure the available head in order to find its horse-power H.P. The head may be measured in feet by aid of a surveyor's level and staff. Then, if w be the weight of a cubic foot of water, and W the weight of the total flow of water per second, we get:—

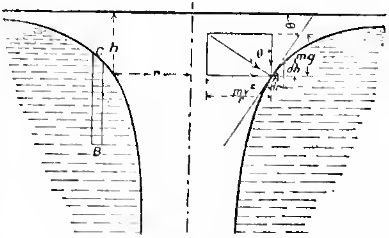
$$Q = a\,v. \quad \text{And } W = Q\,w.$$

$$\text{Hence,} \quad \text{H.P.} = \frac{\text{ft.-lbs. per second}}{550} = \frac{h \times Q\,w}{550}.$$

$$\therefore \quad \text{H.P.} = \frac{62{\cdot}5}{550} \times h\,Q = 0{\cdot}114\,h\,Q = 0{\cdot}114\,h\,a\,v. \qquad \text{(XII)}$$

Vortex Motion.—A whirling mass of fluid is termed a *vortex* and may be either *forced* or *free*. A *free vortex* is one which can be formed naturally, as when water flows through a hole at the bottom of a basin, and is such, that the energy of the fluid per unit mass is the same at all points in it. A *forced vortex* can only be produced artificially, and in it the energy of the fluid is different at different places. Such a vortex may be obtained by rotating a cup containing water.

Free Vortex.—From the above condition of constant energy, the velocity at a point A, on the surface of the vortex and at a depth h below the level of the still water, must be the same as that due to a body falling freely from a height h.



FREE VORTEX.

$$\therefore \qquad v = \sqrt{2\,g\,h}.$$

A particle on the surface of the vortex is acted upon vertically by its own weight $m\,g$, and horizontally by a centrifugal force $\frac{m\,v^2}{r}$. The resultant of these two forces must make an angle θ, with the vertical, so that:—

$$\tan\theta = \frac{\dfrac{m\,v^2}{r}}{m\,g} = \frac{v^2}{g\,r} = \frac{2\,h}{r}.$$

Here m is the mass of the particle and r the radius of the circle in which it revolves.

This resultant must also be perpendicular to the surface of the fluid at A, since the fluid cannot resist any shearing component. Consequently, if the slope of the surface at A be $\tan\theta$,

$$-\frac{d\,h}{d\,r} = \tan\theta = \frac{2\,h}{r}.$$

Or,
$$\frac{d\,h}{2\,h} = -\frac{d\,r}{r},$$

i.e.,
$$\tfrac{1}{2}\log h = -\log r + \log c,$$

$\therefore$
$$\log\sqrt{h} = \log\frac{c}{r},$$

$\therefore$
$$h = \frac{c^2}{r^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (\text{XIII})$$

Here, c is a constant of integration which will depend on the size of the vortex. It is the radius of the vortex at unit depth.

Since the total energy is the same at all places in the fluid and the pressure at B due to the column B C, just makes up for the lower level of B, the velocity in any thin vertical tube of fluid such as B C must be constant and equal to that at the surface of the tube.

A vortex of this kind was applied to centrifugal pumps with radial blades by Professor James Thomson in order to convert, as far as possible, the kinetic energy of the water into potential energy. The vortex commences at the circumference of the wheel and the radial component of the water's motion is outward.

Forced Vortex.—In a vortex whose angular velocity ω is constant throughout, we have at any point A on the surface :—

$$\text{Centrifugal force} = m\,r\,\omega^2,$$

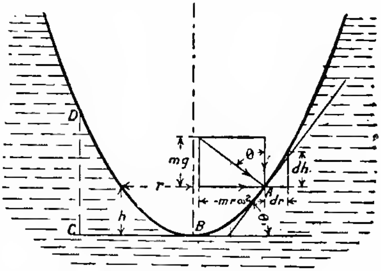
$\therefore$
$$\tan\theta = \frac{m\,r\,\omega^2}{m\,g} = \frac{r\,\omega^2}{g}.$$

$\therefore$
$$\frac{d\,h}{d\,r} = \frac{\omega^2}{g}\,r.$$

Or,
$$d\,h = \frac{\omega^2}{g}\,r\,d\,r.$$

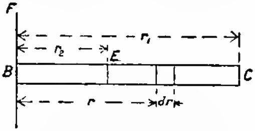
$\therefore$
$$h = \frac{\omega^2\,r^2}{2\,g} = \frac{v^2}{2\,g}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (\text{XIV})$$

This is the equation to a parabola with its axis vertical, and, therefore, the surface of the vortex will be the paraboloid formed by rotating this parabola about its axis. The velocity of any particle D, on the surface of the vortex is that which would be attained by a body falling a distance equal to DC. The kinetic and potential energies are both least at the centre B, and become greater as we move upwards or outwards. We have a vortex of this kind in the wheel of a centrifugal pump with radial blades.



Forced Vortex.

Pressure due to Centrifugal Force.—When a liquid is rotating, its pressure is not the same for all points on one level. Thus, in the previous figure the pressure at B is atmospheric, while at the point C on the same level, the pressure is, in addition, that due to the head C D.



Whirling Column.

Consider a uniform column of liquid B C, rotating round the axis B F, with an angular velocity ω and cross section a. Then, if ρ be the density of the liquid, the centrifugal force due to a small element of length dr, at a distance r from the axis is :—

$$\rho a d r \times \omega^2 r.$$

$\therefore$ The total centrifugal force of the column B C $= \rho a \omega^2 \int r \, d r$.

This force is spread over an area a, at the end C, and we must divide it by this area to get p, the pressure per unit area. If the whole column from B to C is full of liquid :—

Then, $$p = \rho \omega^2 \int_0^{r_1} r \, d r = \tfrac{1}{2} \rho \omega^2 r_1^2 = \tfrac{1}{2} \rho v_1^2 \quad . \quad . \quad . \quad \text{(XV)}$$

But, if the part of the column from B to E is empty :—

Then, $$p = \rho \omega^2 \int_{r_2}^{r_1} r \, d r = \tfrac{1}{2} \rho \omega^2 (r_1^2 - r_2^2) \quad . \quad . \quad . \quad (\text{XV}_a)$$

Students who are not acquainted with the integral calculus will understand the following proof of these equations.

The centre of gravity of B C is at a distance of $\frac{1}{2}$ B C from B.

$$\therefore \left.\begin{array}{l}\text{Centrifugal force due to}\\ \text{the whole column B C}\end{array}\right\} = \rho\, a\, r_1 \times \omega^2 \frac{r_1}{2} = \tfrac{1}{2}\rho\, a\, \omega^2 r_1^2.$$

The centre of gravity of the part E C is distant $\frac{r_1 + r_2}{2}$ from B.

$$\therefore \text{ Centrifugal force due to E C} = \rho\, a\,(r_1 - r_2)\, \omega^2 \frac{r_1 + r_2}{2}$$

$$\text{,, \qquad ,,} \qquad = \tfrac{1}{2}\rho\, a\, \omega^2 (r_1^2 - r_2^2).$$

This formula may be applied to finding the equation for the forced vortex. For, in the figure of the *forced vortex*, if B C $= r$, and C D $= h$, then, $p = h w = h \rho g$.

Therefore, from equation (XV), we get :—

$$h \rho g = \tfrac{1}{2}\rho v^2,$$

Or,

$$h = \frac{v^2}{2g} \text{ (as before).}$$

Reaction of a Jet.—In general, when a fluid issues from an orifice it exerts a force on the vessel which contained it. This force is the reaction of the jet, and is due to the momentum given to the escaping fluid.

Let v = Velocity of efflux, and a = cross-sectional area of outlet.
,, w = Weight of unit volume of the fluid.
,, W = Weight of fluid issuing per second.
,, m = Mass issuing per second.
,, Q = Quantity or volume issuing per second.
,, F = Force exerted on the vessel containing the fluid.

Then, the momentum given to the water per second $= m v$. And, since force is the rate of change of momentum :—

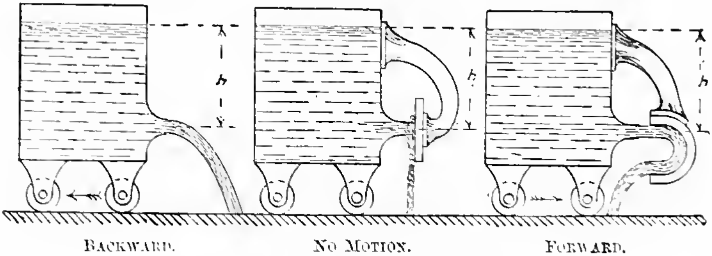
$$F = m v = \frac{W}{g} v.$$

But,

$$W = w Q = w a v.$$

$$\therefore \qquad F = \frac{W}{g} v = \frac{w a v^2}{g} = \frac{w a \,.\, 2 g h}{g} = 2 w a h. \quad \text{(XVI)}$$

This formula also gives the force acting on a vane of a turbine or waterwheel which alters the direction of flow of the water, and in that case, v is the change of velocity produced by the vane. In applying the formula to any particular case, v is the total change of velocity produced until the water is quite clear of the vessel, and the direction of the force acting on the vessel is exactly opposite to that of v. The following figures will explain the results in three cases :—



MOTION PRODUCED BY A JET.

In the left-hand figure the jet of liquid issues towards the right and urges the containing vessel to the left. The jet in the central figure strikes a plate fixed to the tank close to it, and then drops vertically downwards ; consequently, the water receives no horizontal momentum and the tank no motion from it. The water will, however, exert a pressure on the plate, and this pressure balances the force produced by issuing from the orifice. In the remaining figure the jet is turned backwards by a curved guide attached to the tank and the whole momentum imparted to the water is backwards, consequently the tank is pressed forwards. If there be no loss from friction or eddies, the backward velocity of the stream, when the tank is at rest, must be the same as that with which it left the orifice. The blade has not only stopped the water, but has given it an equal backward momentum, and must, therefore, be pressed with a force twice as great as the flat one in the previous case. The force on it, is therefore, 2 F or $4\, w\, a\, h$, which is four times the pressure on a plug stopping the orifice. When the tank is in motion, the momentum given to the water is modified thereby, as will be explained in connection with the Pelton wheel in the next Lecture. Steam life-boats are now frequently propelled by jets of water instead of by screw propellers.

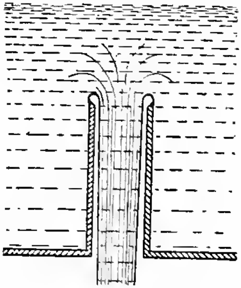
As this reaction astonishes everyone who hears of it for the first time, we will consider it from another point of view. Water under a pressure of 100 lbs. per square inch issues out of a nozzle of 1 square inch in area, and impinges on a fixed vane of such a shape as to gradually deflect the water and turn it back in the opposite direction, without loss by friction, and consequently without loss of velocity. Thus, the vane may be a double **U**, as in the Pelton wheel; or a semicircle; or it may deflect the water in more than one plane. Then, the pressure on the vane is 400 lbs., although the statical pressure on a plug or valve stopping the jet is only 100 lbs. Although it requires very little elementary dynamics to prove this fact, the statement is generally received with incredulity.

In the first place, it will be seen that the vane not only arrests the water—that is, imparts a negative acceleration to it, equal and opposite to what it received from the nozzle—but accelerates it equally in the opposite direction. Therefore, the total pressure on the vane is double the reaction on the nozzle.

The nozzle reaction at first sight appears to be 100 lbs.; that is, area of jet multiplied by pressure. But wait! Imagine a square vertical column in which water is maintained at a constant head. Near the bottom is a suitably formed horizontal nozzle closed by a plug. The system is then in equilibrium, since every square inch of wall has another square inch opposite to it which is acted on by an equal and opposite pressure. Opposite the nozzle, on the other side of the vessel, is a small circle equal to the area of the plug and equally pressed by the water. Now, remove the plug and let the water issue. It really does look as if the force pushing the vessel back was the pressure due to the head acting on this small circle on the back wall—that is, equal to the pressure multiplied by area of jet. But by Newton's second law, whenever we find a body moving at any velocity, we know the product of the force which has urged it and the time during which it has acted. Let us take a simple case and suppose the height of the water surface above the nozzle to be 16 feet. Then we know that the water issues as fast as if it had fallen 16 feet, and that its velocity is 32 feet per second. Consequently, every second a cylinder of water equal in section to the area of the jet and 32 feet long has a velocity of 32 feet per second impressed on it. Therefore, the urging force must be equal to its own weight; because when falling from rest by gravity—that is, when urged by its own weight—it acquires that velocity in one second. But the statical pressure is only half of this, being the weight of a cylinder of water equal in section to the jet and 16 feet long; and, since action and reaction are equal and opposite, the recoil of the vessel is equal to the force urging the jet, or twice the statical pressure.

Of course, if the jet strikes a flat vane at right angles the motion is stopped in its own direction, but not returned, and the pressure on the vane is twice the statical pressure. But, if the vane returns the motion undiminished the pressure is four times the statical pressure.

Reduction of Pressure Round an Orifice.—Let h be the depth of a bell-shaped orifice and a its cross sectional area; then, if this orifice be completely closed by a flat plate, the force required to keep the plate in position will be $w\,a\,h$. It might therefore be supposed that, when the plate is removed and the water allowed to flow, this would be the force exerted on the vessel, but on looking at our former result (see equation XVI) we find that the actual force is twice as great, or $2\,w\,a\,h$. This difference is caused by a diminution of pressure on the surface round about the orifice. At these places the water has a certain velocity, and Bernouilli's theorem shows us that the pressure there must be less than when the liquid was at rest.



RE-ENTRANT ORIFICE.

There can be no reduction of pressure on the flat surface round the bottom of a re-entrant orifice, because there the water is practically at rest. The amount by which the pressure is reduced on opening the orifice must, therefore, be that on the area of the orifice itself, or $w\,a\,h$. Every second, this will set in motion a mass m as given by the equation :—

$$w\,a\,h = m\,v.$$

But,
$$m = \frac{W}{g} = \frac{w\,Q}{g} = \frac{w\,a'\,v}{g}.$$

Where a' is the cross area of the jet, and $v = \sqrt{2\,g\,h}$.

$$\therefore \qquad w\,a\,h = \frac{w\,a'\,v^2}{g} = \frac{w\,a' \times 2\,g\,h}{g}.$$

Hence,
$$a = 2\,a'. \qquad \ldots\ldots\ldots\ldots \qquad (XVII)$$

That is, the area of the jet will only be half that of the orifice. This result has been found experimentally to be correct. With a sharp-edged orifice, such as we considered in the early part of this lecture, there is a certain reduction of pressure round the orifice, and, therefore, the contraction must be less than in this case.

Impact—Loss of Energy.—When a stream of fluid meets an obstruction which causes a sudden change in its motion, its kinetic energy is partially, or perhaps wholly, spent in forming eddies or little whirls of water, and is thus lost so far as useful effects are concerned. The eddies are, however, soon stilled by the viscosity of the liquid and their energy is converted into heat.

Let us suppose a body of mass m_1 moving with a velocity v_1, to overtake and strike another body of mass m_2, and velocity v_2. After collision let the bodies move on together with a common velocity v.

Then, from the laws of momentum,

$$(m_1 + m_2)\, v = m_1 v_1 + m_2 v_2.$$

Or,

$$v = \frac{m_1 v_1 + m_2 v_2}{m_1 + m_2}.$$

But, before impact, the total energy was :—

$$E_1 = m_1 v_1^2 + m_2 v_2^2$$

And, after impact :—

$$E_2 = (m_1 + m_2)\, v^2.$$

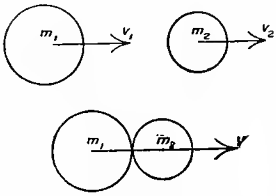
$$\therefore \quad E_2 = (m_1 + m_2) \left\{ \frac{m_1 v_1 + m_2 v_2}{m_1 + m_2} \right\}^2 = \frac{(m_1 v_1 + m_2 v_2)^2}{m_1 + m_2}.$$

The energy lost is :—

$$E_1 - E_2 = \frac{(m_1 v_1^2 + m_2 v_2^2)(m_1 + m_2) - (m_1 v_1 + m_2 v_2)^2}{m_1 + m_2}$$

$$\therefore\ E_1 - E_2 = \frac{m_1 m_2 (v_1^2 + v_2^2 - 2 v_1 v_2)}{m_1 + m_2} = \frac{m_1\, m_2}{m_1 + m_2} (v_1 - v_2)^2. \quad \text{(XVIII)}$$

This lost energy is converted into heat. If one or both of the bodies be fluids the lost energy at first shows itself in eddies; but, as already explained, it is ultimately converted into heat. We thus see that when two streams moving at different velocities mix together, energy is lost and this loss is greater the greater the difference of the velocities. A similar effect takes place when water flows through a pipe with sudden changes of area, and even to a slight extent when the area is gradually varied, and also when water is flowing in a pipe or channel above a certain speed. In all these cases, different parts of the water move with

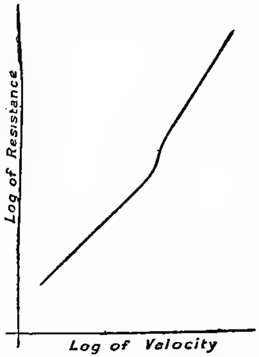


Collision.

different velocities, and these parts get mixed with one another. It also shows, why a turbine or water-wheel in which the water collides with the vanes, must have a low efficiency.*

Resistance of a Pipe.—When a fluid is passing through a pipe it rubs against the sides and experiences a certain resistance to its motion. This resistance limits the flow in a long pipe and causes a loss of head or pressure. Had the water no viscosity its flow would not be affected by the friction of the inner surface; because, this friction could only act on the thin layer of water actually in contact with the pipe.

Professor Reynolds found that the manner in which water flows depends upon its velocity. When this is below a certain critical point the flow depends chiefly on the viscosity and is along smooth stream lines. On passing the critical velocity, the water no longer moves steadily, but breaks up into numberless little whirls or eddies which move along with it, and absorb energy from the main stream. The former condition may be called *Steady Flow* and the latter *Eddy Flow*. This was beautifully demonstrated by Professor Reynolds by passing water from a large tank through a glass tube and then injecting a fine stream of coloured liquid with the same velocity into the centre of the stream. The water in the tank was quite steady and entered the tube through a bell mouth. At low velocities the coloured liquid formed a thin line along the centre of the tube, but at a certain velocity it was seen to suddenly spread out through a considerable part of the water, and on photographing it by means of an electric spark it was found to be all twisted into little whirls.



Resistance of Smooth Lead Pipe at Different Speeds of a Fluid.

He also found, that the law connecting the resistance R, with the velocity v, was different for these two conditions. For the lower speeds he ascertained that the resistance was proportional to the velocity, but at greater speeds it varied as some higher power, ranging from 1·7 to 2 depending on the roughness of the pipe. Near the critical velocity, which depends on the diameter of the pipe and on the temperature, the law is uncertain. The temperature affects the viscosity on which the critical velocity also depends.

On plotting the logarithms of his results as obtained with a smooth lead pipe $\frac{1}{2}$ inch in diameter, we get the lines shown in the accompanying figure. This consists of two straight lines joined by a curve of indefinite shape. The two straight parts have slopes of 1 and 1·72, showing that in one case the increase of log R is equal to the corresponding increment of log *v*, whereas beyond a certain point it is 1·72 times the corresponding increment of log *v*.

---

* See Note 2 at end of this Lecture.

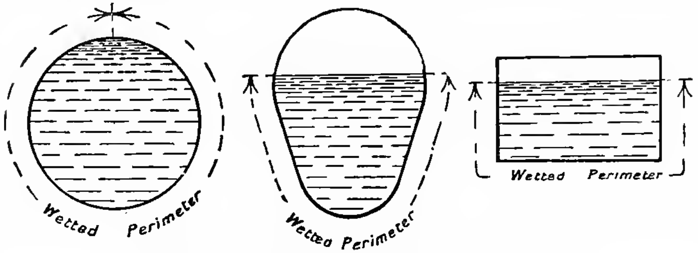
This shows us, that if c_1, c_2, and c_3 be constants, then :—

$$\text{For steady flow, } R = c_1 v.$$

$$\text{For eddy flow, } R = c_2 v^{1\cdot72} + c_3.$$

The resistance of the pipe for any velocity is most conveniently expressed in terms of the difference of pressure per unit length required to force the necessary quantity of liquid per second through it when level. This is called the *Slope of Pressure.* Whether the pipe is level or not, if it be of uniform bore, this slope is given by the difference of level between the free surfaces of the pressure columns at any two points, divided by the length of pipe between them. This has been termed the *Slope of Free Level,* and is shown graphically by a line passing through the surfaces of a series of little pressure columns.

Hydraulic Mean Depth.—The resistance of a pipe or channel is directly proportional to the extent of the wetted surface in a given length, and inversely to the cross area of the stream. The transverse length of the whole surface wetted by the stream is called the *Wetted Perimeter*, and the ratio of the cross area of the stream to its wetted perimeter is termed the *Hydraulic Mean Depth*, which we shall denote by D.



Wetted Perimeter of Different Sections.

The friction of the pipe or channel will consequently vary inversely as the hydraulic mean depth.

Combining this with the former results we have for water pipes or channels :—

$$R = c\frac{v^n}{D}. \qquad \text{(XIX)}$$

Where c is a constant depending on the nature of the surface and the temperature of the water, and n is a number varying from 1·7 for smooth lead pipes to 2 for very rough pipes. The critical velocity for water in ordinary pipes is so low, that in all practical cases in which we wish to know the resistance, the actual velocity is always above it.

For a circular pipe, whose diameter is d, and which is full of running water, the wetted perimeter is the circumference of the circle :—

$$\therefore \qquad D = \frac{\frac{\pi}{4} d^2}{\pi d} = \frac{d}{4} \text{ for a circular pipe.} \quad . \quad . \qquad \text{(XX)}$$

For a rectangular open stream of depth h and breadth b, the wetted perimeter $= 2h + b$ and the area is $b\,h$:—

$$\therefore \qquad D = \frac{b\,h}{2h + b} \text{ for an open rectangle.} \quad . \quad . \qquad \text{(XXI)}$$

Loss of Head due to Friction in Water Pipes.*—Having dealt with the loss of "head" where water enters a pipe due to the shape of its mouth, and mentioned Prof. Reynold's experiments on steady and eddy flows of the liquid through a smooth pipe, as well as explained what is meant by the "wetted perimeter," we are in a position to state the generally accepted rules for the loss of head in ordinary clean cast-iron water pipes.

From many tests with piezometers or pressure columns, gauges or vertical tubes, inserted at known distances apart into the upper sides of straight pipes of different sizes and internal roughness, but of uniform bore, without bends, it has been found that *the loss of head due to friction* for water flowing through these pipes at different velocities, is approximately proportional to—

1. The length of the pipe, l.
2. Inversely to the diameter of the pipe, d.
3. The square of the velocity, v.
4. The roughness.
5. But it is independent of the water pressure.

Let h_T = Total head in feet or pressure per sq. in.
h_L = Lost head due to friction " " "
h_E = Effective head at point of delivery " "

Then, $$h_L = h_T - h_E.$$

---

* This "loss of head due to friction" is termed the "friction head."

Now, if f = coefficient of friction found by experiment, which will vary between 0·012 for a $\frac{1}{2}$-inch pipe when $v = 1$ foot per second, and 0·003 for a 3-foot pipe when $v = 15$ feet per second.

Then, loss of head,

$$h_L = f\left(\frac{\text{wetted surface}}{\text{cross area}} \times \text{loss of head due to velocity}\right).$$

$$\text{Or,}\quad h_L = f\left(\frac{\pi\, d\, l}{\frac{\pi}{4} d^2} \times \frac{v^2}{2\, g}\right).$$

$$\therefore\quad h_L = f\left(\frac{4\, l\, v^2}{d\, 2\, g}\right). \quad \ldots\ldots\ldots \quad \text{(XXII)}$$

Example II.—What will be the loss in head for every 100 feet of a 3-inch pipe when water flows through it with a velocity of 3 feet per second, if the coefficient of friction be ·0065 ?

Here, $l = 100$ feet; $d = 3$ inches $= ·25$ feet; $v = 3$ feet per second; and $f = ·0065$.

By formula XXII., Answer—

$$\text{We see that,}\quad h_L = f\left(\frac{4\, l\, v^2}{d\, 2\, g}\right).$$

$$\therefore\quad h_L = ·0065\left(\frac{4 \times 100 \times 3 \times 3}{·25 \times 2 \times 32·2}\right).$$

$$\text{Or,}\quad h_L = 1·46 \text{ feet.}$$

Now, looking at the following table by Prof. Merriman, we see that opposite to a pipe of 0·25 foot in diameter, and directly under the velocity 3 feet per second, the loss of head is printed as 1·46 feet for every 100 feet of length. In that table, the "friction factor," f_F, is reckoned as four times the previously mentioned value for the coefficient of friction, f.

$$\text{Or,}\quad f_F = 4\, f;$$

because the constant 4 appears in the numerator of the natural equation XXII.

Hence, when using the table, the "friction factors," f_F, will vary from 0·05 for a $\frac{1}{2}$-inch pipe when $v = 1$ foot per second, to 0·012 for a 3-foot pipe when $v = 15$ feet per second.

The formula for the "friction head" therefore becomes, using the following table:—

$$h_L = 4\, f\left(\frac{l\, v^2}{2\, d\, g}\right) = f_F\left(\frac{l\, v^2}{2\, d\, g}\right). \quad \ldots \quad \text{(XXIII)}$$

TABLE OF FRICTION HEAD FOR 100 FEET OF CLEAN IRON PIPE.*

Diameter of Pipe.	Velocity in Feet per Second.						
	1.	2.	3.	4.	6.	10.	15.
Feet.	Feet.	Feet.	Feet.	Feet.	Feet.	Feet.	Feet.
0·05	1·46	5·10	10·30	16·90	34·70	...	...
0·10	0·59	1·99	4·20	6·97	14·50	37·30	...
0·25	0·20	0·70	1·46	2·40	5·37	13·70	29·40
0·50	0·09	0·32	0·70	1·14	2·46	6·22	13·30
0·75	0·05	0·21	0·45	0·73	1·57	3·94	8·40
1·00	0·04	0·15	0·32	0·55	1·12	2·80	5·95
1·25	0·03	0·11	0·25	0·42	0·85	2·11	4·48
1·50	0·02	0·09	0·20	0·33	0·67	1·66	3·50
1·75	0·02	0·07	0·16	0·26	0·54	1·33	2·80
2·00	0·02	0·06	0·13	0·21	0·45	1·09	2·27
2·50	0·01	0·05	0·10	0·16	0·34	0·81	1·68
3·00	0·01	0·04	0·07	0·12	0·26	0·67	1·40
3·50	0·01	0·03	0·06	0·10	0·21	0·53	...
4·00	...	0·02	0·05	0·08	0·17	0·42	...
5·00	...	0·02	0·04	0·06	0·13	...	...
6·00	...	0·01	0·03	0·05	0·10	...	...

EXAMPLE III.—Find the friction in a 4-inch wrought-iron pipe delivering 200 gallons per minute. What is the loss in head and H.P. per 100 feet? (See *The Practical Engineer*, Sept. 11, 1903, p. 264.)

ANSWER.—Gallons per second $= \frac{200}{60}$.

Cubic feet per second $= \frac{200 \times 10}{60 \times 62{\cdot}3} = {\cdot}535.$

Area in sq. ft. of a 4″ pipe $= {\cdot}087.$

Velocity (v) of water $= \frac{{\cdot}535}{{\cdot}087} = 6{\cdot}1$ ft. per sec.

Let the friction factor $= {\cdot}023.$

Then, by formula XXIII. and the above table—

We get, $$h_{\mathrm{L}} = f_{\mathrm{F}} \left(\frac{l\,v^2}{2\,d\,g}\right) = {\cdot}023 \left(\frac{100 \times 6{\cdot}1^2}{2 \times {\cdot}3 \times 32{\cdot}2}\right) = 4 \text{ ft.}$$

H.P. absorbed $= \frac{4 \times 200 \times 10}{33{,}000} = {\cdot}24$ (approximately).

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* From *Treatise on Hydraulics*, 8th Edition, by Prof. Merriman, Lehigh University.

Example IV.—A hydraulic lift, with ram, load, &c., weighs 10 tons, the ram is 9 inches in diameter, and the friction in the mechanism is equal to 0·05 of the gross load. The accumulator is half a mile away, and is loaded to 800 lbs. per square inch, the diameter of the supply pipe is 3 inches. Estimate the speed of ascent of the lift, if the loss of head in the pipe is $0{\cdot}0005\, l v^2/d$; l being the length in feet, d the diameter in feet, v the velocity of water in the pipe in feet per second. (B. of E., S. 3, 1904.)

Answer.—Let V be the speed of the lift in feet per second. Then, since the diameter of the supply pipe is $\frac{1}{3}$ the diameter of the ram, the speed of water in the supply pipe will be equal to 9 V. Assuming that the "loss of head" is measured in feet of water, we get :—

$$\left.\begin{array}{r}\text{The loss of pressure in}\\ \text{lbs. per square inch}\end{array}\right\} = {\cdot}43 \times \frac{0{\cdot}0005\, l\, v^2}{d} \left\{\begin{array}{l}\text{Since } {\cdot}43 \text{ lb. per}\\ \quad\text{sq. in.} = 1 \text{ foot}\\ \quad\text{head of water.}\end{array}\right.$$

$$\text{,,} \qquad \text{,,} \qquad = \frac{{\cdot}43 \times 0{\cdot}0005 \times 2{,}640 \times (9\text{ V})^2}{{\cdot}25}$$

$$\text{,,} \qquad \text{,,} \qquad = 184\text{ V}^2.$$

Thus, the pressure on the ram will be equal to $(800 - 184\text{ V}^2)$ lbs. per square inch.

Equating the work done per second by the water on the ram to the work per second required to raise the lift, &c., we have—

$$\left.\begin{array}{l}\text{Pressure per sq. inch}\\ \quad\text{on the ram} \times \text{area of}\\ \quad\text{ram} \times \text{its speed}\end{array}\right\} = \left\{\begin{array}{l}\text{(Weight of lift + loss due to its}\\ \quad\text{friction)} \times \text{its speed.}\end{array}\right.$$

$$(800 - 184\text{ V}^2) \times \frac{\pi \times 9^2}{4} \times \text{V} = (10 \times 2{,}240 + {\cdot}05 \times 10 \times 2{,}240) \times \text{V}.$$

$$\text{,,} \qquad \text{,,} \qquad = 10 \times 2{,}240(1 + {\cdot}05) \times \text{V}.$$

$$\text{Hence,} \qquad \text{V}^2 = \frac{800}{184} - \frac{10 \times 2{,}240 \times 1{\cdot}05 \times 4}{\pi \times 9^2 \times 184}.$$

$$\text{Or,} \qquad \text{V}^2 = 4{\cdot}35 - 2{\cdot}01 = 2{\cdot}34.$$

$$\therefore \qquad \text{V} = \sqrt{2{\cdot}34} = 1{\cdot}53 \text{ feet per second.}$$

NOTE ON WORKING OF HYDRAULIC RAMS (see p. 121).

Professor Irving P. Church in his text-book on *Hydraulic Motors* gives the following as the mode of action of the hydraulic ram :—

"The water in the drive-pipe having thus been brought to rest, but being still slightly compressed, an elastic recoil or rebound takes place and the pressure underneath the two valves C V and D V quickly falls to a low value, less than one atmosphere, so that the valve C V closes, while D V is opened, both on account of its weight and of *the pressure of the outer air*.

"RESULTS OF EXPERIMENTS ON A NO. 2 GOULD'S HYDRAULIC RAM, carried out by Professor Church at Cornell University in March, 1899.

No.	Whole Time in Minutes	Pulsations per Minute.	Length of Stroke of Waste Valve in Closing.	Lbs. of Water Raised per Minute.	Height (H) through which Water is Raised, in Feet.	Lbs. of Water Passing through Waste Valve per Minute.	Fall or Working Head (h) of Waste Water, in Feet.	Efficiency of Ram. η.
1	12	80	$\frac{7}{8}''$	5·50	49·4	26·7	17	0·598
2 (*s*)	15	66	$\frac{7}{8}''$	2·30	56·4	21·8	10	0·595
3 (*s*)	16·7	98	$\frac{5}{8}''$	4·82	49·4	20·6	17	0·680

"In No. 1 the full weight of the waste valve D V, which was $6\frac{5}{8}$ ounces, was operative, there being no provision to counterpoise a portion of it; and the length of its movement, or 'stroke,' was $\frac{7}{8}$ inch. In the other two, Nos. 2 (*s*) and 3 (*s*), a light spring was employed, by the use of which the waste valve was virtually relieved of about one-half of its weight (though, of course, its 'mass' was practically unchanged). In No. 2 (*s*), the length of stroke of valve being the same as before, the efficiency is maintained at nearly 60 per cent., notwithstanding the fact that the ratio $H \div h$ is nearly double what it was in No. 1. This is doubtless due to the (relatively) quick closing of the valve. In No. 3 (*s*) the stroke has been made shorter with the effect of increasing the efficiency to 68 per cent., to which the decrease in the ratio $H \div h$ has also probably contributed somewhat. The pulsation is here very rapid, being 98 to the minute."

NOTES ON MEASUREMENT OF A FLOWING STREAM (see p. 129).*

Note 1. **Ritchie's Rule for the Flow of a Stream with a Rectangular Notch.**—"Cube the depth of the water in inches, flowing through the notch, extract square root, multiply by 5 (or more exactly by 4·83)—this gives the number of cubic feet per minute flowing over each foot in width."

Or, $$Q = k\,b\,h^{\frac{3}{2}} = \left(5 \times 1 \times h^{\frac{3}{2}}\right).$$

"I have found the best proportion of width of notch to depth flowing over the notch to be about 10 to 1."

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* I am indebted to Mr. J. Ritchie, of Carrick & Ritchie, Engineers, Waverley Engineering Works, Edinburgh, for these *Notes*.

Note 2, *re* p. 141, at top.—Mr. J. Ritchie says: "I am not quite clear about this. As a matter of fact, the turbine buckets at the point of entry are nearly always at right angles, or square on to the jet; especially, in all the mixed flow turbines, which are rapidly superseding the older forms of Jonval and other Continental makes. Most of the Continental turbine makers are now taking up the manufacture of mixed flow turbines. In Girard turbines, it is usual to have the outer edge of the bucket slightly curved in the direction of the flow, but the circumferential speed of a Girard turbine is generally lower in proportion to the velocity of the water, than a pressure turbine of the Jonval, inward, or mixed flow type. Some of these Girard turbines give the best results at ·45 of the theoretical velocity due to the head, whereas the usual proportion of a pressure turbine is ·65 of the theoretical velocity due to the head. Some of the buckets of American wheels are actually rounded on the face, instead of hollow, such as the Risdon turbine, and these give very high efficiencies. In some of the very latest American turbines the vanes are perfectly radial in plan, the turning action on the water being entirely done on the lower part of the bucket. In these cases, the water issues from the port almost at right angles to the bucket, so that even allowing for the forward motion of the wheel there must be collision to some extent."

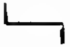
Lecture IV.—Questions.

1. Prove the law for changing p, v, and h along a stream line in a frictionless fluid. Apply the law to find the funnel shape of the surface of water in a basin from which the water is flowing by a central hole.

2. Prove the law for changing p, v, and h along the stream lines in a frictionless fluid. Apply, neglecting change of level, to the case of adiabatic flow of air from one vessel to another through a small orifice, and deduce the rule for maximum quantity flowing.

3. Describe the action of a jet pump, or of a good form of injector.
4. Water flows through a round sharp edged orifice 3 inches in diameter in a flat plate, at about 12 inches below still-water level. Show by a sketch your notion of the shapes of the stream lines. If we wish to know the pressure at any point, why is it not sufficient to know only the depth?

5. Deduce a formula giving the velocity with which water issues from an orifice, and show how to apply it to water under pressure.
6. Discuss briefly the relative advantages under various circumstances of the different methods of measuring a stream of water.
7. Find the horse-power of a waterfall 70 feet high, when the stream is such that it passes over a gauge notch 6 feet long with a head of 15 inches. If it is employed to drive a turbine of 80 per cent. efficiency, what B.H.P. would you expect to obtain?
8. Investigate the form of the surface of water which flows out of a hole in the bottom of a basin with a vortex motion.
9. What is meant by a constrained vortex, and why will such a vortex rapidly disappear when left to itself? Find the form taken by the surface and show how to apply this to finding the pressure in a centrifugal pump.
10. The wheel of a centrifugal pump 2 feet outside diameter has a very large case and rotates at 100 revolutions per minute about a vertical axis, and almost no water is being delivered. Calculate and show in a curve the pressure at points in a horizontal plane, at various distances from the axis. The vanes are bent backwards at an angle of 30° with the radius at the outer part; if the radial flow becomes 2 feet per second and the circumferential openings are 200 sq. inches in area, what is the kinetic energy of the water leaving the wheel? What is the pressure in excess of that at the inner rim of the wheel where the water enters without shock? If there is no frictional loss to what height will the water be lifted above the well? In an actual pump with small wheel case what is the probable lift?

11. "Barker's mill" consists of a horizontal pipe with a nozzle at right angles to it at each end thus  Water enters it by a vertical pipe at the centre, and the whole is so mounted that it can rotate about the axis of this pipe. Find the torque when the water is issuing under a head h, and the mill is revolving n times per minute. Show that the power is greatest when the velocity of the nozzles is half that due to the head, and that the efficiency then cannot be over 50 per cent.
12. What are the chief conclusions to be drawn from Reynold's experiments on the flow of water through pipes?
13. What is meant by the *Hydraulic Mean Depth* of a pipe or channel, and show how we use it in calculating the resistance to the flow of water?

14. If a hydraulic company supplies 1,000 gallons of water at 700 lbs. pressure per square inch for 17 pence, how much is this per horse-power per hour? Another company supplies water at a "head" of 260 feet: what price ought it to charge per 1,000 gallons if the consumer is to have his energy at the same price as in the other case? *Ans.* 2·08d.; 2¾d. (B. of E. Adv., 1900.)

15. What is the usually accepted rule for loss of head or energy per pound of water passing along a straight pipe? What is the law when the water flows very slowly? And why is there a difference between the two laws? (B. of E. Adv., 1900.)

16. 60 horse-power is being delivered hydraulically into a mile of straight horizontal pipe at a pressure of 700 lbs. per square inch. It is found that 52 horse-power is available at the far end. What power will be found available at the end of 2 miles of a pipe three-fourths of the diameter of the first-named, if the entering pressure is 800 lbs. per square inch, and the entering horse-power is 40? *Ans.* 26·61 H.P. (B. of E. H., Part I., 1900.)

17. A horizontal pipe of 12 inches diameter gradually becomes 3 inches diameter, and then becomes of 12 inches diameter again. There is a flow of 5 cubic feet per second. Neglecting friction, state how the pressure alters along the axis of the pipe. (B. of E. Adv., 1901.)

18. Take the loss of energy through friction of every pound of water flowing along a straight circular pipe of length l feet and diameter d feet, the velocity being v feet per second, to be $0{\cdot}0007\ l v^2/d$ foot-pounds. If water at 700 lbs. per square inch conveying 300 horse-power enters a pipe of 6 inches diameter, find the number of cubic feet entering per second and the velocity in the pipe. What is its loss of energy per pound in a length of pipe of 3,000 feet? What is the total loss of power? Obtain an expression for the horse-power lost in transmission in terms of the total horse-power entering the pipe, the entering pressure, p, and l and d. (B. of E. Adv. & H., Part I., 1902.)

19. Eight gallons of water per second flow through a 6-inch pipe in which there is a right-angled bend. State the speed. What is the change in the velocity of the water (that is, the *vector* change, as there is no change in mere speed)? What is the change in the momentum of the water per second? What is the resultant force exerted by the water on the pipe at the bend, neglecting friction? (B. of E. Adv. & H., Part I., 1902.)

20. A stream is gauged over a rectangular weir or notch, the width of the notch is 3 feet 6 inches, and the height of the still water over the edge of the notch is found to be 14¼ inches. Find how many gallons pass over the weir per 24 hours (the velocity of approach may be neglected), and what H.P. could be obtained from turbines supplied by this stream if the available head is 35 feet and the mechanical efficiency of the turbines is 81 per cent. (C. & G., 1900, H., Sec. C.)

21. Compare the loss of head by skin friction in a 3-inch pipe with that in a 6-inch pipe, when the velocity of flow is the same in both, and also when the velocity in the small pipe is so increased that it discharges as much water as the big pipe. (C. & G., 1900, H., Sec. C.)

22. Water is flowing steadily at a velocity of 3 feet a second through a perfectly horizontal pipe 6 inches in internal diameter; if on a length of 1½ miles there is a drop of pressure owing to friction of 25 lbs. per square inch, what H.P. is being wasted in friction? (C. & G., 1900, O., Sec. A.)

23. A large tank, in which a constant head of water is maintained, has a cylindrical hole in the bottom 3 inches in diameter, with sharp edges. How many gallons of water will escape from the tank per hour if the constant head is 9 feet? (C. & G., 1901, H., Sec. C.)

24. Obtain an expression for the loss of head, owing to surface friction, in a long straight pipe through which water is flowing. Hence obtain an expression for the probable discharge in gallons per hour from a main of given diameter and length, when the total head of the water in the reservoir above the point of discharge is known.
(C. & G., 1901, H., Sec. C.)

25. The quantity of condensing water used by a stationary engine is gauged by first allowing the water to flow into a tank supplied with baffle plates, in order to get rid of any agitation in the water, and then allowing it to flow out of the tank over a rectangular notch 6 inches wide. Estimate the discharge in pounds per minute when the head over the sill is 4 inches. Explain why the flow may be more accurately determined by a V-notch than by a rectangular notch, the head over the notch being supposed subjected to variation. (C. & G., 1902, H., Sec. C.)

26. A horizontal water pipe, 3 feet in diameter, gradually contracts to a diameter of 18 inches. The difference of head, as shown by water gauges, between two sections of these diameters is 28 inches. Neglecting all losses, estimate the flow along the pipe in gallons per hour.
(C. & G., 1902, H., Sec. C.)

N.B.—*See Appendices B and C for other questions and answers.*

Lecture IV.—A M.Inst.C.E. Exam. Questions.

1. Explain what is meant by coefficients of contraction, velocity, and discharge for orifices. What are the ordinary values of these coefficients for a sharp-edged orifice in a plane surface? (I.C.E., *Oct.*, 1897.)

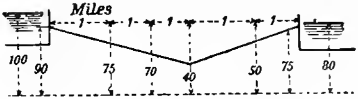
2. Deduce an expression for the discharge from a rectangular notch, and explain why the theoretical expression requires to be modified by a coefficient varying with the proportions of the notch. (I.C.E., *Oct.*, 1897.)

3. A weir is 30 feet long, and has 18 inches of head above the crest. Taking the coefficient at 0·6, find the discharge in cubic feet per second. State the relative advantages of notches with and without end contraction in the practical gauging of water. (I.C.E., *Oct.*, 1897.)

4. A pipe of 12 inches diameter connects two reservoirs 5 miles apart, and having a difference of level of 20 feet. Find the velocity of flow and discharge of the pipe, taking the coefficient of friction at any value known to you, or at 0·01. Explain how the coefficient of friction for a pipe varies with roughness and size. (I.C.E., *Oct.*, 1897.)

5. Describe carefully any one method of determining the discharge of a river by float or current-meter observations. Show roughly by a sketch how the velocities are distributed in the cross-section of a rectangular channel. Describe any instruments used in determining the velocity, in the method adopted in your description. (I.C.E., *Oct.*, 1897.)

6. Explain the meaning of the term hydraulic gradient of a pipe. Suppose the levels of the reservoirs and pipe at mile distances are 100, 90, 75, 70, 40, 50, 75 and 80 feet above datum, write down the corresponding pressures in the pipe in feet of head. (I.C.E., *Oct.*, 1897.)



7. On what principle is the presence of a jet striking a plane determined? Find the pressure of a 3-inch jet having a velocity of 80 feet per second against a wall struck normally. (I.C.E., *Oct.*, 1897.)

8. A jet of water 1 inch in diameter coming from a reservoir at a height of 200 feet strikes a fixed hemispherical cup so that the direction of its motion is reversed. Find the force it exerts upon the cup, assuming that the jet has 90 per cent. of the full velocity due to the head.
(I.C.E., *Feb.*, 1898.)

9. The miner's inch is defined as the flow through an orifice in a vertical plane, of 1 square inch in area, under an average head of $6\frac{1}{2}$ inches. Find the water-supply per hour which this represents, and give any reasons you can for the choice of coefficients which you assume. (I.C.E., *Feb.*, 1898.)

10. A canal lock with vertical sides is emptied through a sluice in the tail-gates. Putting Ω for area of lock-basin, ω for area of sluice, and H for the lift, find an expression for the time of emptying the lock.
(I.C.E., *Feb.*, 1898.)

11. A re-entrant cylindrical mouthpiece is fixed to the vertical side of a vessel. Show that for such a mouthpiece the limiting value of the coefficient of contraction is 0·5. (I.C.E., *Feb.*, 1898.)

12. A rectangular weir, for discharging daily 10 million gallons of compensation water, is arranged for a normal head over the crest of 15 inches. Find the length of the weir. Assume your own coefficient, stating the kind of weir for which it is applicable, or take a coefficient of 0·7.
(I.C.E., *Feb.*, 1898.)

13. A water-main is 42 inches in diameter, and the velocity through it is to be 3 feet per second. Find the head lost in friction in feet per mile. Coefficient of friction 0·01. (I.C.E., *Feb.*, 1898.)

14. A horizontal main consists of 5,000 feet of 6-inch, 5,000 feet of 9-inch, and 5,000 feet of 12-inch diameter. The velocity in the 6-inch is 12 feet per second, and the pressure head 50 feet at the inlet. Sketch (not to scale) the hydraulic gradient (*a*) supposing that only the skin friction is considered; (*b*) supposing that both skin friction and the losses of head due to abrupt changes of section are considered. Write down the expressions for the several losses of head, but without calculating them numerically.
(I.C.E., *Feb.*, 1898.)

15. Deduce a general expression for the velocity of flow in an open channel. An irrigation channel, with side slopes at 1 to 1, has a bottom width of 100 feet and a depth of 10 feet. It discharges 3,000 cubic feet per second. Find the slope in feet per mile. (I.C.E., *Feb.*, 1898.)

16. Write down a formula for the discharge of pipes, and explain its derivation. What coefficients would you use to calculate the discharge for (1) pipes coated with Dr. A. Smith's composition, (2) riveted steel pipes, (3) wooden-stave pipes. (I.C.E., *Oct.*, 1898.)

17. A pipe 2 feet diameter draws water from a reservoir at a level of 550 feet above the datum; it falls for a certain distance and again rises to a level of 500 feet, 5 miles from its starting point; it then falls to a reservoir at a level of 400 feet 1 mile away. Calculate the rate of delivery into the lower reservoir. (I.C.E., *Oct.*, 1898.)

18. Describe one form of current meter, and state how you would proceed to gauge a river 100 feet wide and 20 feet deep with it. (I.C.E., *Oct.*, 1898.)

19. Give an expression for the loss of head in a pipe owing to a sudden enlargement of section. (I.C.E., *Oct.*, 1898.)

20. Describe the hydraulic ram, and give a formula for its delivery at a height h feet above it, the supply being Q cubic feet per second coming from a height of H feet. (I.C.E., *Oct.*, 1898.)

21. State Bernoulli's theorem, and explain what form of motors could be used to utilise each of the forms in which energy exists in flowing water.
(I.C.E., *Oct.*, 1898.)

22. Give a formula for the discharge of drowned orifices. What advantages do they offer for gauging purposes? (I.C.E., *Oct.*, 1898.)

23. A pipe tapers to one-tenth its original area, and then widens out again to its former size. Calculate the reduction of pressure, at the neck, of the water flowing through it, in terms of the area of the pipe and the velocity of the water. Why is this reduction of pressure a gauge of the discharge? (I.C.E., *Oct.*, 1898.)

24. Sketch a hook gauge for measuring a varying water-level, and state how you would fix it for use. (I.C.E., *Feb.*, 1899.)

25. Water flows from a pond over a weir 10 feet long, to a depth of 10 inches; it then flows along a level rectangular channel 8 feet broad, and over a second weir the width of the channel, its crest being 1 foot above the bottom. Find the depth of the water over the 8-foot weir.
(I.C.E., *Feb.*, 1899.)

26. A rectangular chamber 120 feet square contains 15 feet depth of water, which is allowed to flow out through a vertical rectangular orifice

2 feet by 1 foot, the top of which is level with the floor of the reservoir and the tail-water. Calculate the time it will take to empty.

(I.C.E., *Feb.*, 1899.)

27. How would you proceed to gauge accurately, by means of floats, the flow of a stream of uniform section 20 feet wide and 5 feet deep? Describe the floats you would use, and state how you would deduce the result from the experimental data. (I.C.E., *Feb.*, 1899.)

28. Calculate what depth of water should flow in a pipe of circular bore to give a maximum discharge, the surface slope being constant.

(I.C.E., *Feb.*, 1899.)

29. (*a*) Explain by sketches the phenomenon of contraction in the case of a weir with vertical face and ends and with a sharp edge such as is commonly employed for measuring the discharge of water, and illustrate in reference to the same sketches the meaning of the terms "coefficient of contraction" and "coefficient of discharge." (*b*) What is approximately the coefficient of discharge where the weir is 10 feet long and the water 1 foot deep? Would this coefficient be affected by reducing the length of the weir to 1 foot while maintaining the same depth of water, and if so, in what manner? (I.C.E., *Oct.*, 1899.)

30. Explain and illustrate by reference to a stream of rectangular section, 12 feet wide and 3 feet deep, the term "hydraulic mean depth," and state in what manner and approximately to what extent the mean velocity of the stream would be affected by widening it to 24 feet, assuming the depth and inclination to remain the same. (I.C.E., *Oct.*, 1899.)

31. In the same stream as above (12 feet wide and 3 feet deep), in what manner and to what extent would the mean velocity be affected by changing the inclination from 1 per 1,000 to 4 per 1,000, assuming the depth to remain the same? (I.C.E., *Oct.*, 1899.)

32. Sketch a practical form of hydraulic ram for raising automatically to a considerable height a portion of any volume of water derived from a lesser height, and explain the points upon which satisfactory working depends. (I.C.E., *Oct.*, 1899.)

33. A certain tank A overflows with the discharge from a 12-inch pipe fed by a tank B 10 miles distant and 90 feet above A. The discharge is 900,000 gallons a day. Another 12-inch pipe, 5 miles long, is subsequently joined to the middle distance of the 10-mile length and discharges into a third tank C having its overflow at the same level as A. State approximately the discharge per day into A and C under the new conditions.

(I.C.E., *Oct.*, 1899.)

34. A stream of water of uniform rectangular section, 12 feet wide and 3 feet deep, has a surface inclination of 1 foot per 5,280 feet. Assuming the channel to be of rough masonry making the coefficient of friction 0·007, what would be the mean velocity in feet per second and the discharge in cubic feet per second? (I.C.E., *Oct.*, 1899.)

35. A large cistern is divided vertically by a partition into two compartments A and B. In the partition is a square orifice. Water is maintained at a constant level, in compartment A, of $1\frac{1}{2}$ feet, and in compartment B of 1 foot, above the centre of the orifice, and the discharge from A to B through the orifice proves to be at the rate of 0·2 cubic foot per second. (*a*) Assuming the same orifice to be 6 inches lower and the water-levels and all other things to be unchanged, what now will be the discharge through the orifice from A to B? (*b*) State approximately in decimals of a foot the depth and width of the orifice, assuming it to be cut square in a thin plate. (*c*) Would the discharge of the orifice be affected by making its width three times as great and its depth one-third as great? If so, how and why? (I.C.E., *Feb.*, 1900.)

36. Assuming that in the following pairs, A and B, of straight streams the roughness of the sides is moderate and uniform, state in the simplest way possible whether No. 1 or No. 2 would discharge more water, and by how much per cent.

A.

	No. 1.	No. 2.
Length,	1 mile.	1 mile.
Fall,	4 feet.	6 feet.
Top width,	70 ,,	50 ,,
Bottom width,	70 ,,	50 ,,
Depth,	10 ,,	10 ,,

B.

Length,	1 mile.	1 mile.
Fall,	4 feet.	4 feet.
Top width,	70 ,,	60 ,,
Bottom width,	50 ,,	60 ,,
Depth,	11 ,,	10 ,,

If in B the coefficient of friction were 0·007, what would be the mean velocity? (I.C.E., *Feb.*, 1900.)

37. Draw approximately to a scale of 1 inch to the foot and foot-second a diagram of depths and corresponding velocities parallel to the axis, and extending from the bottom to the free surface, in the centre line of a river 10 feet deep, when the mean of such velocities is 5 feet per second; and explain the causes of variation of velocity from the bottom upwards, with special reference to the velocity at the free surface. (I.C.E., *Feb.*, 1900.)

38. A straight thin cylindrical tube, 0·25 foot internal diameter, is fixed horizontally through the vertical side of a cistern, its outer end being flush with the outside thereof. The inner end is cut square and projects into the cistern as far as is consistent with allowing a jet of water issuing through it from the cistern to contract freely in air without striking the interior of the tube. (*a*) Assuming the level of the water in the cistern to be 2 feet above the centre of the tube, sketch such an arrangement in outline approximately to a scale of 3 inches to 1 foot, showing the figure of the jet, and figuring the principal dimensions. (*b*) What is the distance from the cistern to an ideal vertical plane which the centre line of the jet would cross at a level of 1 foot below the level of water in the cistern? (I.C.E., *Feb.*, 1900.)

39. Describe shortly, with sketches approximately to scale, the different classes of instruments used for measuring the velocity of open streams of water, explaining the circumstances under which each is to be preferred. (I.C.E., *Feb.*, 1900.)

40. A jet of water having a sectional area of 12 square inches and a velocity of 16 feet per second impinges normally on a plane surface. Find the amount of work per second necessary to maintain the jet and determine the pressure on the plane, (*a*) when it is fixed, and (*b*) when it is moving in the direction of the jet at 6 feet per second. (I.C.E., *Oct.*, 1900.)

41. What is the connection between the velocity and pressure at different parts of a smooth horizontal pipe of varying diameter and which is running full of water, all resistance being neglected? If the pressure where the diameter is 2 inches is 500 lbs. per square inch and velocity 4 feet per second, what will be the pressure where the diameter is 1 inch? (I.C.E., *Oct.*, 1900.)

42. In a pipe 2 inches inside diameter and tarred inside, water, starting at 700 lbs. per square inch, flows at the rate of $2\frac{1}{2}$ feet per second. What is the horse-power transmitted at the beginning of the pipe, and what percentage of this is lost on friction and viscosity per 100 feet length of pipe? (I.C.E., *Oct.*, 1900.)

43. If the same horse-power with the same initial pressure as in (42) were transmitted through a $1\frac{1}{2}$-inch pipe, how much would this percentage loss on friction be increased, and what would be the saving in weight of metal pipe per 100 feet length? Make a rough allowance for weight of joint flanges in this comparison. (I.C.E., *Oct.*, 1900.)

44. Explain, without mathematical symbolism, the difference between the water-resistances to the progress of a vessel through open deep water and through a canal. (I.C.E., *Oct.*, 1900.)

45. Calculate the percentage loss of H.P. per 1,000 yards in hydraulic transmission through clean cast-iron pipes $4\frac{1}{2}$ inches in diameter, with water velocities 2 feet, 3 feet, and 4 feet per second, and with pressures 200 lbs., 700 lbs., and 1,200 lbs. per square inch, that is, in all 9 percentage losses, and represent your calculated results by curves drawn on squared paper. (I.C.E., *Feb.*, 1901.)

46. A measuring weir is constructed with a 90° angular notch, the edges being bevelled to 45° on the outside to a nearly sharp edge. Give the formula you think best for the discharge over such a weir, and apply it to calculate the discharge in gallons per minute when the water depth above the apex of the angular notch is 9·36 inches and the water level 5 feet back from the weir is found to be 0·93 inch above that of the weir. (I.C.E., *Feb.*, 1901.)

47. Define "hydraulic mean depth" and calculate its amount for a pipe 4 feet in diameter flowing full, the same pipe half-full, and for a channel 4 feet bottom width, side slopes 2 to 1, and a depth of water 4 feet. Also, find the rate of flow in the pipe when full, and in the channel. The hydraulic gradient in the former case, and the slope in the latter case being 1 in 3,000, assuming an approximate value for the coefficient. (I.C.E., *Oct.*, 1901.)

48. Develop a formula for the flow over a rectangular notch; if the velocity of approach is appreciable, how may it be allowed for in the calculation? (I.C.E., *Oct.*, 1901.)

49. Explain how the velocity at different points in the cross section of a river varies, and show how this variation is inconsistent with stream-line motion. (I.C.E., *Oct.*, 1901.)

50. A circular orifice in the bottom of a tank is 0·385 inch diameter, and the mean head of water over the orifice is 22 inches. Calculate the theoretical flow of the water through the orifice, assuming the velocity to be due to the "head" only. What would be the energy of one pound of this water? (I.C.E., *Feb.*, 1902.)

51. Water flows through a straight pipe 4 feet in diameter and 100 feet long. The upper end is 10 feet higher than the lower end. At the middle of its length there is a conical contraction in the pipe, after which it is again enlarged to its original diameter. The velocity of the water through the full-sized section is 5 feet per second. Find what the diameter of the contracted section must be for the pressure there to be 10 lbs. per square inch less than at the upper end, neglecting friction. (I.C.E., *Feb.*, 1902.)

52. Give a formula for the resistance to the flow of water (1) in a smooth tube where the water moves with low velocities; (2) in a pipe where the water flows at high velocities. Find the loss of head due to friction in water flowing through a pipe 2 feet in diameter, 1 mile long, at 5 feet per second. (I.C.E., *Feb.*, 1902.)

53. Find the quantity of water which will flow through a notch 9 feet long, the head of water over the sill being 10 inches, and the area of the approach channel being 30 square feet. (I.C.E., *Feb.*, 1902.)

54. Describe one form of current meter, and explain how it could be used to ascertain the discharge of a river 50 feet wide. (I.C.E., *Feb.*, 1902.)

55. Find an expression for the pressure on a plate due to the impact of a jet of water. What difference would result if the plate were replaced by a hemispherical bowl? Water is flowing through a pipe 80 feet long at the rate of 50 feet per second, and is stopped by a valve being closed in $\frac{1}{10}$th of a second. What is the increase in pressure near the valve, the pipe being supposed rigid and with no elasticity? (I.C.E., *Oct.*, 1902.)

56. State and prove "Bernoulli's Theorem" relating to the steady motion of incompressible fluids, and apply it to find the discharge through a Venturi water-meter. (I.C.E., *Oct.*, 1902.)

57. Estimate the discharge of a clean cast-iron pipe 15 inches in internal diameter, 2 miles long, under a total head of 100 feet. Give reasons for the selection of the formula you use, and state what you neglect in your calculations. (I.C.E., *Oct.*, 1902.)

58. Obtain an expression for the loss of head due to a sudden enlargement in a pipe, and calculate the loss in the case of a pipe 2 inches in internal diameter suddenly enlarged to 4 inches, the discharge being 5000 gallons per hour. (I.C.E., *Oct.*, 1902.)

59. Show that, for a rectangular notch, a formula of the form $Q = a(L - bh)h^{\frac{3}{2}}$ can be made to represent the discharge at varying heights more correctly than a formula of the form $Q = c\,L\,h^{\frac{3}{2}}$. (I.C.E., *Oct.*, 1902.)

60. A jet of water, velocity 24 feet per second, diameter 3 inches, strikes a plane normally. Find, approximately, the total pressure on the plane and the maximum pressure, and show by a curve the approximate distribution of pressure. (I.C.E., *Oct.*, 1902.)

61. Show, by sketches relating to any case you may select, how the velocity varies at different points in the cross-section of a stream. (I.C.E., *Oct.*, 1902.)

62. Describe and discuss (*a*) a rough method, (*b*) a more accurate method, of estimating the discharge of a stream. (I.C.E., *Oct.*, 1902.)

63. Water issues from the nozzle of a fire-engine, 1½ inch diameter, in a jet which rises to a height of 100 feet. Neglecting the contraction of the jet, find the reaction on the machine due to the jet. (I.C.E., *Feb.*, 1903.)

64. A horizontal tube is tapered slowly from a diameter of 15 inches to a diameter of 6 inches. Neglecting friction, calculate the difference in the pressures in pounds per square inch at the two sections when the discharge is 60,000 gallons per hour. (I.C.E., *Feb.*, 1903.)

65. In the formula $v = c\sqrt{m\,i}$ for the mean velocity in streams and channels, explain how the value of c is affected by the roughness of the sides and bottom, the slope, and the hydraulic mean depth. Selecting a suitable value of c, find the discharge in cubic feet per second of a rectangular rough masonry channel 5 feet wide, when the depth of the water is 2 feet and the slope is 1 in 250. (I.C.E., *Feb.*, 1903.)

66. What are the advantages and disadvantages attending the use of the V-gauge notch, and for what purpose is it specially suitable? The still-water surface-level is at a height of 15·5 inches above the bottom of a right-angled V-gauge notch. Calculate the discharge in cubic feet per second, taking 0·60 as the coefficient of discharge. (I.C.E., *Feb.*, 1903.)

67. Sketch carefully a form of current meter. Explain exactly how you would employ it, and from its indications obtain the mean velocity of a river. (I.C.E., *Feb.*, 1903.)

68. At a point in a clean hydraulic supply main, 3 inches in diameter, the velocity is 10 feet per second, and the pressure is 750 lbs. per square inch; find the percentage loss of pressure per 1,000 yards, and calculate the power conveyed by the main at the point. (I.C.E., *Feb.*, 1903.)

69. Give a brief account of Reynolds' experiments on the flow of water through pipes and the results obtained from them. (I.C.E., *Feb.*, 1903.)

70. Describe and explain the action of a hydraulic ram, noting the chief points which must be attended to in its design and erection.
(I.C.E., *Feb.*, 1903.)

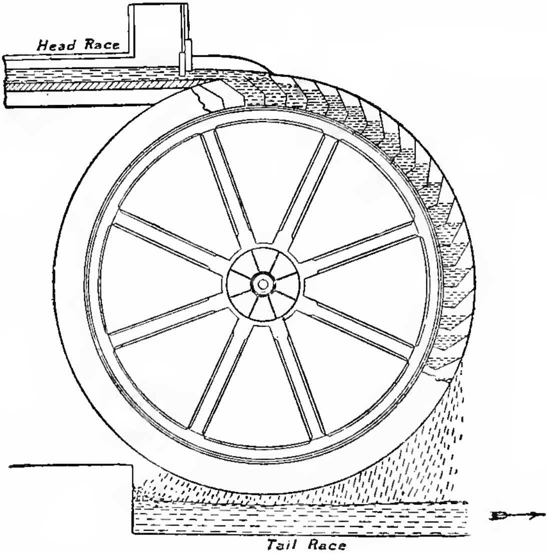
N.B.—*See Appendices B and C for other questions and answers.*

LECTURE V

WATER-WHEELS AND TURBINES.

CONTENTS.—Hydraulic Motors—Overshot Water-wheel—Breast-wheels—Undershot Water-wheel—Fairbairn's Improvements—Clack Mill—Pelton Wheel—Turbines—Girard Turbine—Jonval Turbine—Günther's Governor—Thomson's Vortex Turbine—Little Giant Turbine—Hercules Mixed-Flow Turbine—Centrifugal Pumps and Fans—Notes on this Lecture—Questions.

Hydraulic Motors.*—In connection with hydrostatics we have already described some machines for obtaining motion by hydraulic means, but in all these cases the water acts solely by its pressure,



OVERSHOT WATER-WHEEL.

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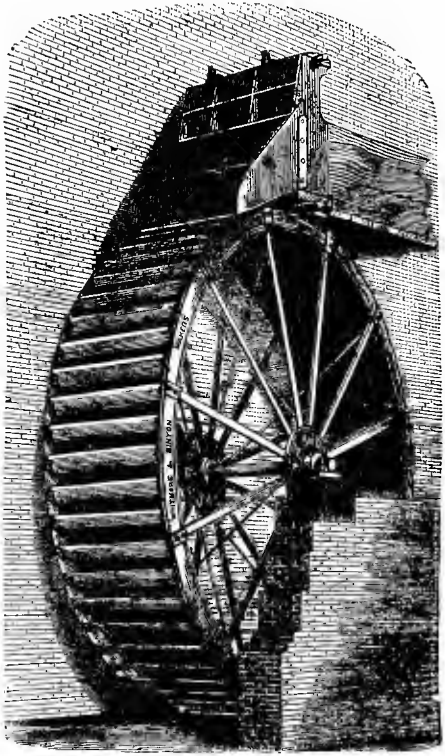
* Students should refer to *Hydraulic Motors* by G. R. Bodmer (London, Whittaker & Co.), and to *Hydraulic Machinery* by R. Gordon Blaine (London, E. & F. N. Spon, Limited) for further information on the design of water-wheels and turbines.

We now come to the consideration of other water motors in which the weight and momentum of the water are also employed. These may be divided roughly into two classes—Water-wheels, in which the water acts for the most part directly by its weight; and turbines, in which it acts by its momentum. We cannot, however, draw a very sharp line between them, as they gradually merge into one another, and the power of both ultimately depends on gravity.

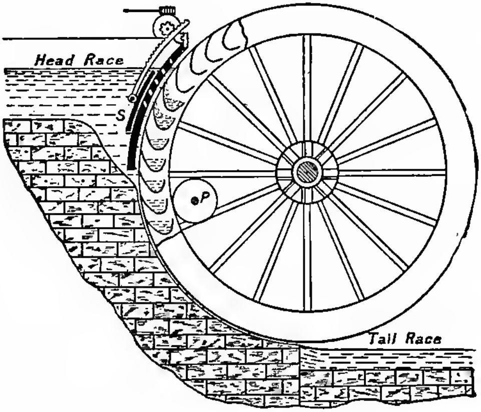
Overshot Water-wheel.—This consists of a wooden or iron frame to which is fixed a number of blades, so as to form with the inner circumference a series of buckets for holding water. The water is led along an aqueduct termed the *head race* to the top of the wheel, and there enters the buckets. Its weight forces them downwards and thus makes the wheel revolve. As each bucket in turn approaches its lowest position the water gradually drops out of it into the tail race. The motion of the water as it enters the wheel also assists to some extent in producing rotation. The buckets are so shaped and fixed to the wheel, that as little as possible of the available power shall be lost by the water spilling from them before they reach the tail race. The sectional view shows one form of bucket for this purpose whilst the outside view shows another with curved blades. One disadvantage of this wheel is, that the water leaves



OVERSHOT WATER-WHEEL BY MESSRS. WHITMORE & BINYON.

it with a velocity opposite in direction to that of the tail race. if this flows the same way as the head race. The water therefore does not get away so freely from the tail race, and more clearance is necessary at the bottom of the wheel which thereby involves a loss of head.

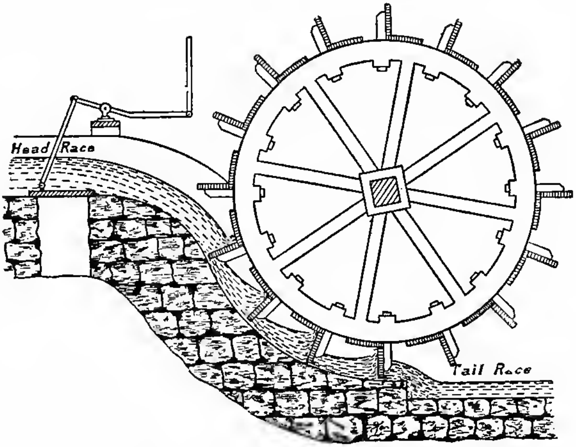
Breast-wheels.—This last consideration, coupled with the difficulty of supporting the head race for large overshot wheels, has brought about the introduction of breast-wheels, in which the water is introduced between the top and middle of the wheel as shown by the next illustration. The breast-wheel is also frequently made with curved blades into which the water drops almost vertically, and then acts chiefly by its weight. The motion of the wheel in this case assists the escape of the tail water instead of hindering it as in the previous one. The curved ventilated form of bucket with closed breast, as shown in the above figure, was first introduced by Sir William Fairbairn and greatly increased the efficiency of the motor. But, for small wheels, such as are used for farms, where first cost is more important than efficiency, hey are usually made radial as shown by the following figure.



FAIRBAIRN'S BREAST-WHEEL.

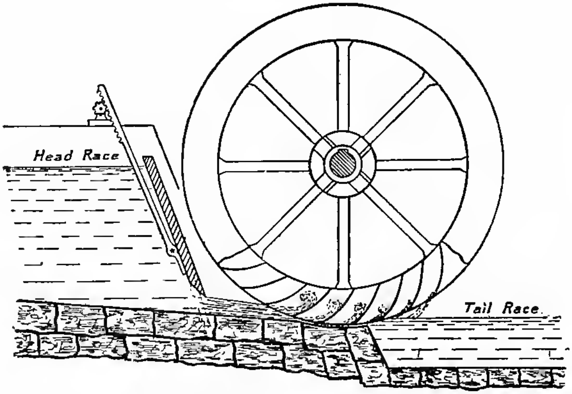
Here, the water is allowed to attain a certain amount of motion before reaching the wheel, and therefore acts partly by momentum and partly by its weight. The buckets have no ends, but the wooden breast serves to keep the water from escaping by the sides and circumference of the wheel before it reaches the bottom.

Breast-wheels into which the water enters near the top are called high breast-wheels.



BREAST-WHEEL WITH RADIAL BLADES AS USED IN COUNTRY FARMS.

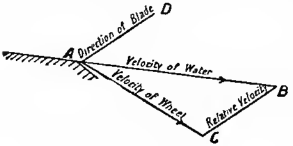
Undershot Water-wheel.—The breast-wheel just described forms a connecting link between water-wheels proper and turbines, to which latter class undershot wheels really belong. With an undershot wheel the whole energy of the water is allowed to become kinetic and then it acts on the blades solely by its momentum. Here,



UNDERSHOT WATER-WHEEL.

again, radial blades are common, but they are very inefficient on account of the large loss due to the eddies which are caused by the impact of the water upon them, and from the kinetic energy which the water still possesses on leaving the wheel.

In order that the water may be applied without shock and to avoid the formation of eddies, the tips of the blades should be so inclined as to be parallel to the motion of the water relatively to the wheel at its point of entering. To find this direction, draw A B to represent the velocity of the water as it leaves the head race, and A C the circumferential velocity of the wheel at the point where the water enters it. Then, by completing the triangle A B C we find C B the direction of the tip of the blade at A. The blade is curved upwards, and the position where the water leaves it is at the same level as where it enters, in order that the water may drop out with as little kinetic energy as possible.



Angle of Blade.

Fairbairn's Improvements.—In the illustration of Fairbairn's breast-wheel, we have shown the regulating sluice S connected by a curved rack and pinion to the worm gear which is turned by a hand-wheel, until the sluice admits the desired quantity of water to develop the necessary power. In the case of large wheels which have to drive textile or other machinery requiring great uniformity of speed, this worm gear is connected to a ball governor which automatically adjusts the position of the regulating sluice, to suit the different demands of the works.

In addition to other improvements effected by Fairbairn, we may mention, that instead of driving from a spur-wheel keyed to the water-wheel shaft, he bolted a segmental annular toothed wheel directly to one of its outer sides or flanges and geared it with a pinion as shown at P. He thus diminished the distance between the plummer block bearings of the water-wheel shaft, relieved its radial arms from conveying the driving stresses, and at once obtained the necessary speed without intermediate gearing. The importance of this improvement was so self-evident to mill-owners, that many large wheels which had previously given considerable trouble and shown signs of distress, were fitted with Fairbairn's drive, and are working to the present day at full power with perfect regularity and freedom from breakdowns.

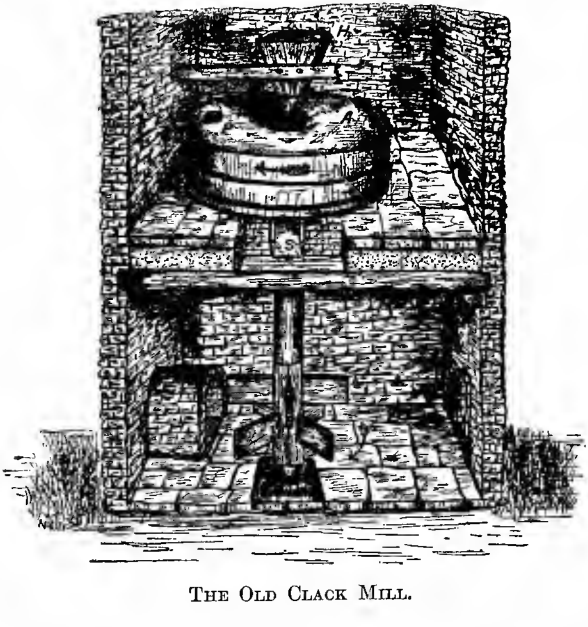
We may here summarise Fairbairn's improvements in water-wheels, viz. :—(1) The use of iron instead of wood in their construction, thus lightening and at the same time strengthening the various parts.

(2) The adoption of curved ventilated iron buckets instead of straight wooden ones, to prevent eddies and thus obtain a greater efficiency from the head and body of water.

(3) The introduction of a closed breast to prevent the escape of the water during its turning of the wheel.

(4) Driving directly from the sides and periphery of the water-wheel in order to minimise the stresses in the arms, reduce the distance between the bearings, and at once obtain the desired speed.

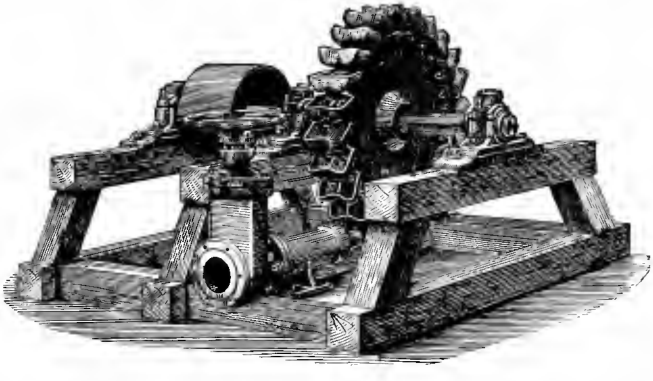
The Clack Mill.—One of the oldest forms of water-wheels, and one which belongs to the same class as the undershot wheel, is that known as the "clack mill." It is supposed to be of Norwegian origin, and was certainly introduced therefrom into the Orkney and Shetland Islands several hundred years ago. Our illustration is reduced from a sketch made by Mr. Sellar in 1898 of the only remaining working clack mill in Orkney, at the farm of Millbrig, in the parish of Birsay. The owner, Mr. Folster, says that it has been handed down to him from generation to generation for more than two hundred years.



THE OLD CLACK MILL.

The water under a head of 10 feet enters by the left-hand aperture and leaves by the right-hand one. During its passage the water impinges upon the further side of the wooden radial arms which are fixed to the vertical iron shaft. The lower end of this shaft rests in a conical footstep, whilst the upper end carries the revolving millstone, and is steadied by a bearing immediately beneath the lower or fixed stone. The corn to be ground is fed into a "head" or "harp" H, and as the upper stone revolves the projecting wooden pin P, strikes the radial arm A connected to the lower end of H. This shoggles a portion of the grain into the central opening at each revolution. In doing so, a clacking noise is made by this pin striking the outstanding arm, which has given rise to the local term of "clack mill." The grain is carried down by gravity and finds its way into groves between the two stones where it is ground into rough meal. This compound of meal and husk dribbles from the shoot S into a wooden bin, from which it is removed, and separated by shaking and blowing, or by another machine, for future use in the shape of porridge or oat-cake.

Pelton Wheel.*—The previous examples naturally lead us to the consideration of the Pelton wheel, which is very often used for great pressures and high falls. As will readily be understood from a consideration of the accompanying figure, this form of water-wheel consists of a plain disc mounted upon a central shaft, and carrying a number of curved buckets fixed at equal distances around its periphery. A conical nozzle attached to the supply pipe is so fixed as to direct a jet of water upon each of these buckets in turn and thus drive the wheel at a high speed. The



PELTON WATER-WHEEL BY MESSRS. W. GÜNTHER & SONS, OLDHAM.

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* See Notes on this water motor at end of this Lecture.

buckets have a central division, and as they curve outward towards each side, the jet is thereby deflected in two portions and then backwards. With properly designed buckets, and when the circumferential velocity of the wheel is half that of the jet, the water will simply leave the buckets with little or no remaining kinetic energy, and hence the efficiency of such a wheel may be very great. Pelton wheels are used for falls having a head of from 30 to 2,000 feet, or for corresponding pressures derived from water-pumps and city hydraulic power mains. They are sometimes made with several nozzles, each being fitted with a stop-valve, so that the power can be varied by shutting off the water from one or more of them. When there is only one jet, the power can be varied without a change of efficiency by simply unscrewing the nozzle and putting on another of a different size. If the power is varied by partially closing the stop valve, we lose a large amount of energy in friction at the valve, and the efficiency is thereby considerably reduced.

We may find the efficiency of a Pelton wheel, on the assumption that there are no eddies or friction, as follows :—

Let V = The velocity of the water as it issues from the nozzle.
,, v = ,, ,, vane.

Then, $V - v$ is the relative velocity of water on vane, which is not changed as it moves around the vane. Therefore, the final velocity of the water is $v - (V - v)$ or $2v - V$, which may be either positive or negative. Now, by the principle of conservation of energy, the initial energy of the water is equal to that spent on the vane + the final energy. Therefore, the efficiency of the wheel is :—

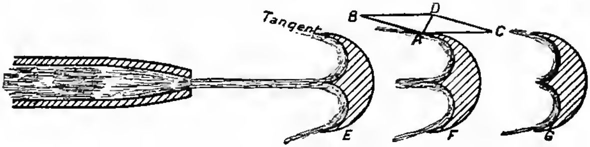
$$\eta = \frac{\textit{Power got out}}{\textit{Power put in}} = \frac{\textit{Initial kinetic energy} - \textit{final kinetic energy}}{\textit{Initial kinetic energy}}$$

$$= \frac{V^2 - (2v - V)^2}{V^2} = \frac{4\,v\,(V - v)}{V^2}.$$

This is evidently greatest, and equal to unity, when $2v - V$ is zero, or $V = 2v$, since $(2v - V)^2$ can never be negative. That is, for maximum efficiency the speed of the wheel should be half that of the water, and when it runs either quicker or slower than this the efficiency will be less. For example, if the speed of the jet is 100 feet per second and the vanes travel 20 feet per second, the theoretical maximum efficiency is :—

$$\eta = \frac{4 \times 20\,(100 - 20)}{100 \times 100} = 0{\cdot}64 \text{ or } 64 \text{ per cent.}$$

The above are ideal cases in which the water is directed backward by the vane parallel to its original direction. In practice, the Pelton vanes do not quite do this, but are slightly divergent, as shown, in order that the water may clear the following vane.



VANES FOR PELTON WHEEL.*

The final velocity of the water is easily found by the parallelogram of velocities, thus—Draw A B the velocity of the water and A C that of the vane. Then, A D is the final velocity of the water, and the kinetic energy carried away is proportional to A D².

We have seen, in the preceding Lecture, that when the ideal vane is stationary, the pressure on it is four times the statical pressure on the area of the jet. At maximum efficiency, when the relative velocity of the water and vane is half what it is when the vane is stationary, the force on the vane is equal to this statical pressure, because the impact varies as the square of the relative velocity. Or,

Water striking vane per second $= \frac{w}{g} a (V - v)$ *lbs.*

Change of velocity produced by vane $= 2 (V - v)$.

∴ Momentum imparted per second $= \frac{w}{g} a (V - v) \times 2 (V - v)$.

Or, Force of jet on vane . . . . . $= \frac{2 w a}{g} (V - v)^2$.

And, when, $v = \frac{1}{2} V = \frac{1}{2} \sqrt{2 g h}$,

Then, $F = \frac{2 w a}{g} (\frac{1}{2} \sqrt{2 g h})^2 = w a h$.

Hence, in this case, the power expended on the vane is $F \times \frac{1}{2} v$. But, the total power of the jet is $F \times v$, or twice as much. This would make the efficiency to be one half, but we have proved it to be unity. Where is the discrepancy?

Suppose a single vane to move away indefinitely in a straight line in the direction of the jet, it will be seen that there is an ever-increasing quantity of water flying through the air after the vane. This represents an ever-increasing debt of uncollected kinetic energy, so that only half the water issuing per second strikes the vane in the same time.

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* See Notes at end of this Lecture.

But now, let E be the initial position of the vane (see previous figure), and E F the distance it travels in one second. When it has arrived at F, let another vane be suddenly interposed at E. There is now a rod of water between E and F running twice as fast as the vanes, and the last particle of this water will only overtake F when it arrives at G and the second vane at F. A third vane is now interposed at E and the process repeated.

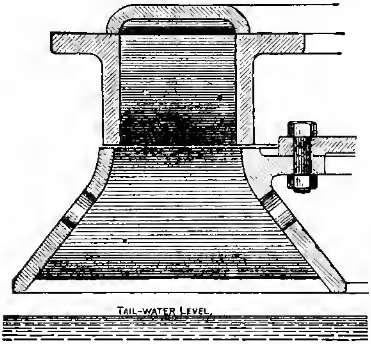
This is what happens in the Pelton and other impact wheels, such as the Laval steam turbine. The fluid strikes two vanes at once, one behind the other; a statement hard to believe unless approached by the above argument.

Turbines.—These form a type of water motor which occupy much less space, are more efficient, more easily governed, suit a greater range of fall, and generally run at a greater speed than ordinary water-wheels. They are classified in several different ways according to the manner in which their special properties are considered. We shall first of all divide them into four classes, viz.—(1) inward flow; (2) outward flow; (3) parallel (or axial) flow; and (4) mixed flow turbines. In the first two kinds, the water either flows inwards from the outer circumference as in the Thomson turbine, or outwards from the inner circumference as in one type of the Girard. In the third class, it flows parallel to the axis, entering at one side of the wheel and leaving at the other as in the Jonval. The fourth type has both radial and axial flow, and the water usually enters at the circumference and leaves at one or both sides parallel to the axis.

In the second place, we find all of the previous types divided into what are termed *drowned* and *ventilated* turbines. The former are designed to run quite full of water and may be submerged in the tail race, or used with a suction pipe; whilst the latter have ducts to admit air at atmospheric pressure into the wheel, and, consequently, cannot be submerged or used with such a pipe. The energy of the water entering the drowned type is partially potential and partially kinetic, whilst it is wholly kinetic before it enters a ventilated turbine. Hence, these two kinds of motors are sometimes respectively called *reaction* and *impulse* turbines.

The purpose of a suction pipe is to allow the turbine to be placed some distance above the tail race without losing the corresponding head. This suction pipe is merely an air-tight conduit to carry away the used water and must have its lower end below the surface of the tail race; moreover, it must not exceed 20 feet or thereby in vertical height from the turbine to the tail race; otherwise the water column supported by the pressure of the atmosphere will be broken by the introduction of air into the exhaust conduit.

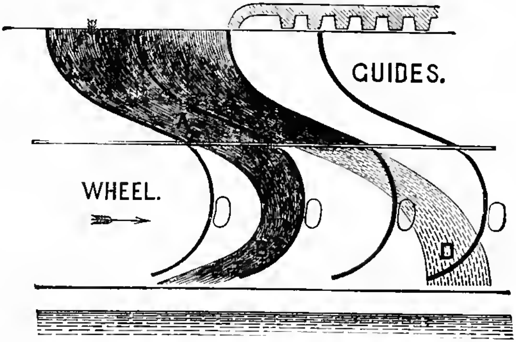
A disadvantage of most turbines of the "drowned" type is, that in regulating its speed and power we cannot gradually cut off the water without a considerable drop in efficiency. We can, however, do so in several steps when divisions are placed in the wheel for this purpose. This is because the guide passages to the wheel must always be quite full of water, which would not be the case if the opening to any particular guide port was only partially closed.



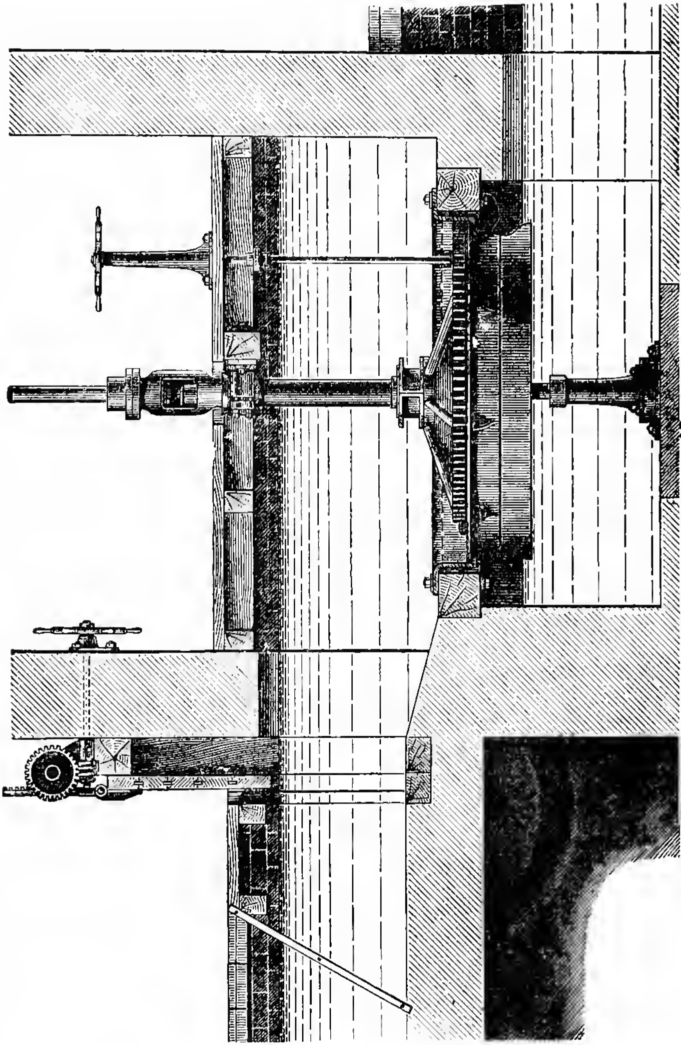
Radial Section. Günther's Axial Flow Girard Turbine.

Girard Turbine.—This wheel is named after the French engineer M. Girard, who in 1856 first designed the ventilated and impulse turbine. It can be made with either a vertical or horizontal axis and for both axial and outward flow. The former is used for low falls of from 6 feet and upwards, and the latter for high falls up to 1,000 feet.

By referring to the two accompanying figures, which represent radial and circumferential sections of this turbine, it will be seen that the gate for controlling the water supply is placed above the guide ports A, C, through which the water passes and issues with full velocity due to its

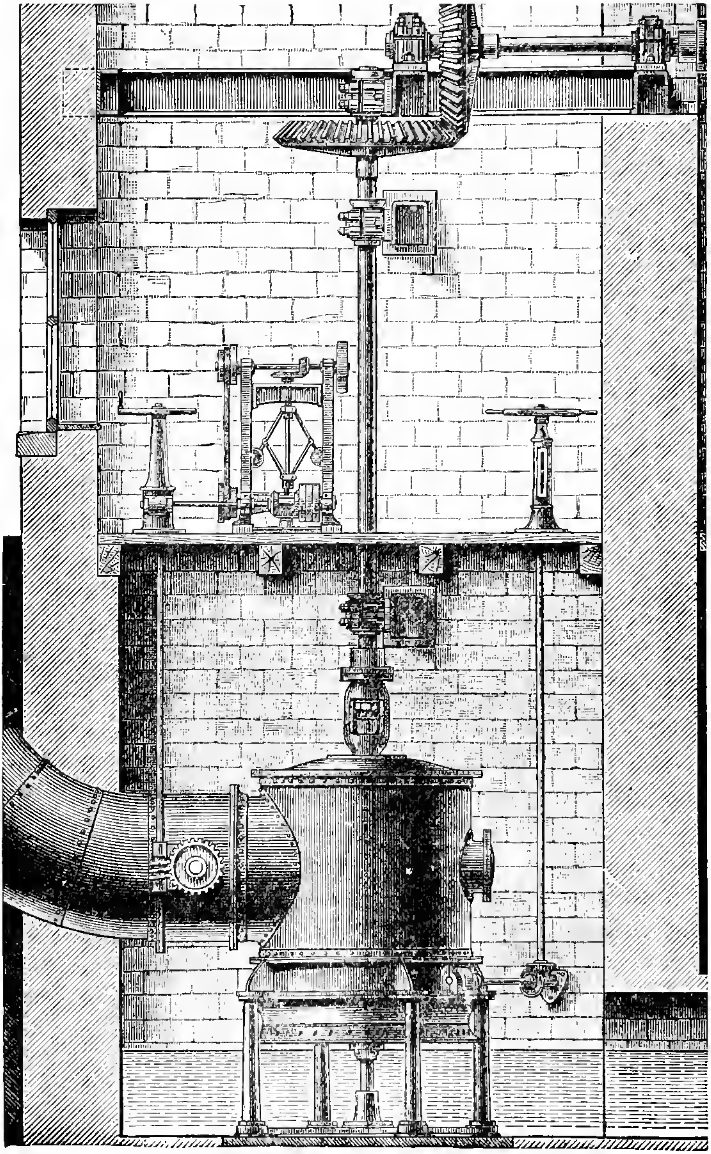


Circumferential Section. Günther's Axial Flow Girard Turbine.



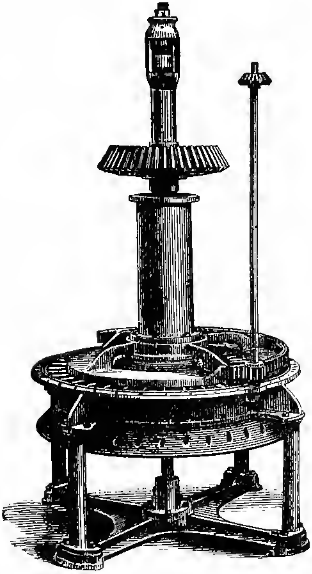
Girard Turbine for Low Falls by Messrs. W. Günther & Sons, Oldham.

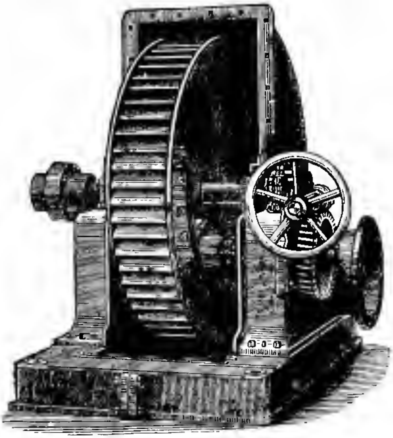
head. It then glides along the concave surfaces of the wheel buckets B, but does not quite fill them. The inclination of the upper edges of the buckets is obtained in the manner explained



Günther's Girard Turbine for Low Falls.

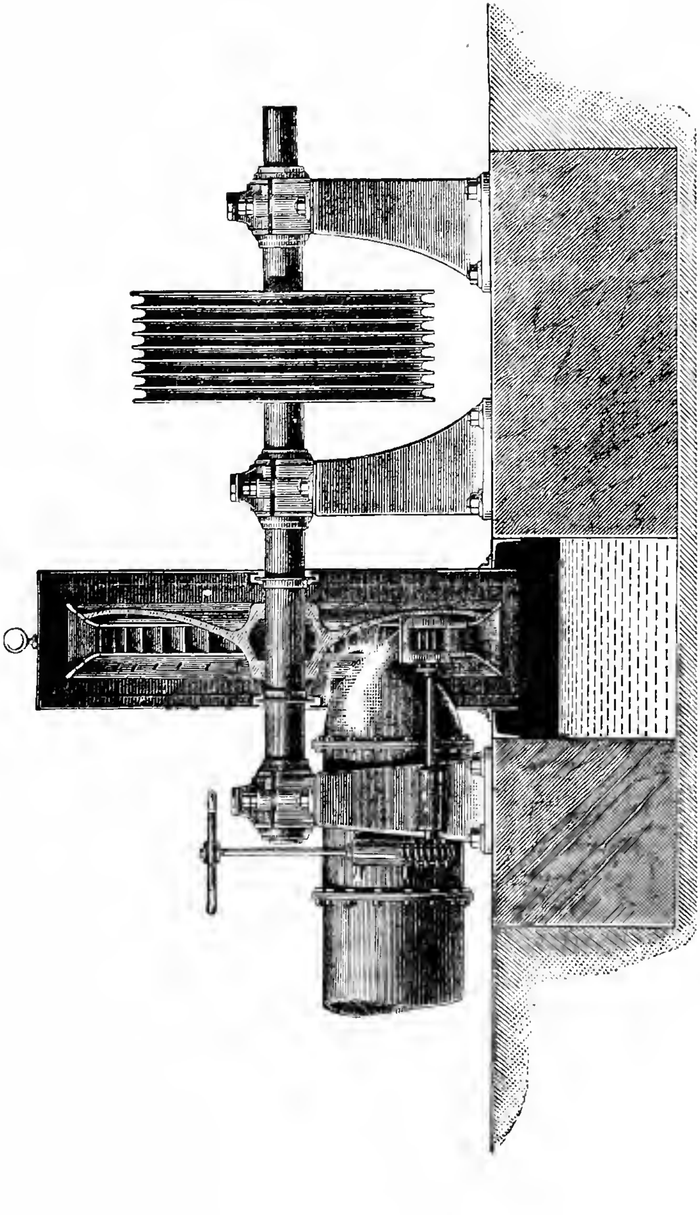
for undershot wheels, while that of the lower edges is made as small as possible in order that the water may not leave the vanes with much kinetic energy. To allow of this inclination being smaller than would otherwise be the case, the sides of the wheel are splayed outwards as shown by the radial section in order that the water may spread and not foul the convex surface of the next blade. The path of the water relatively to the moving wheel is shown at B, whereas, the dotted lines through D show the actual motion of the stream as seen from a fixed point. The ventilating holes are clearly shown in both views. These admit air to the wheel and prevent the formation of eddies in the empty parts.


GÜNTHER'S GIRARD TURBINE FOR LOW FALLS.


GÜNTHER'S GIRARD TURBINE FOR HIGH FALLS.

The above left-hand illustration is that of a complete turbine, and the two full-page illustrations show how they are fixed in position ready for work. The second of these also indicates how the governor is attached.

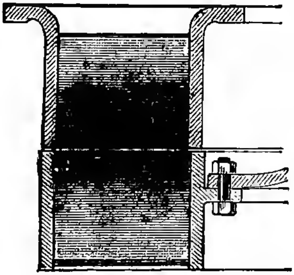
When used for high falls the water is only admitted to a few of the buckets at a time. It is then usually made with outward



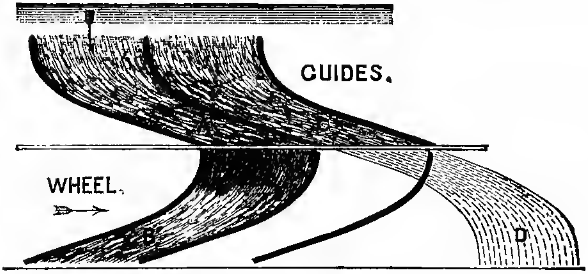
Longitudinal Section of Günther's Girard Turbine for High Falls.

flow, and mounted on a horizontal axis as illustrated by the sectional drawing facing this and the right-hand figure on the previous page. The pipe for admitting the water and the hand gear for adjusting the flow are clearly visible, whilst a grooved pulley for rope driving is placed on the right. This type of turbine gives a high efficiency at medium as well as at full load.

Jonval Turbine.—This turbine is of the parallel or axial flow type and is designed to be always kept full of water. The accompanying figures give radial and circumferential sections, having the same lettering as the corresponding figures for the Girard turbine. It is usually employed for low and medium falls of from 2 to 40 feet, and can run equally well when completely submerged or when connected to a suction pipe. The adjustment of the water supply is effected by means of a slide or slides which close the guide passages one after another. A turbine of this type has the greatest efficiency when all its passages are full of water, and consequently the size of the wheel depends on the quantity of water to be passed in a given time. The reason why this class of turbine is not used for small power with high falls is, that very little water would be required, and, therefore, the dimensions of the wheel would be very small —perhaps impracticably small—whilst the speed of rotation would be very great. With the Girard type on the other hand, the efficiency is not reduced by having only one or two of the



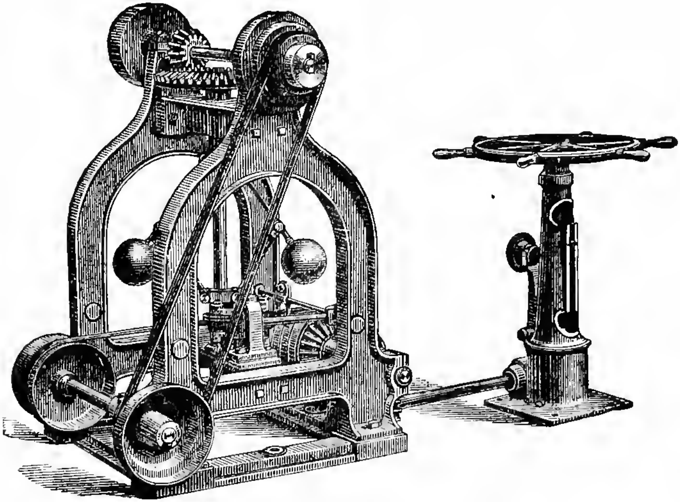
Radial Section. Günther's Jonval Turbine.



Circumferential Section. Günther's Jonval Turbine.

buckets in use at one time. We can, therefore, employ as large a wheel as we like, and only use the requisite number of buckets for the required power. This enables us to get a slower speed of rotation which is usually desirable. The Girard type, however, cannot work with a suction tube, and only works well when clear of the tail water. An inch or two of fall, must, therefore, be sacrificed, and this reduces the efficiency with low falls.

Günther's Turbine Governor.—For many purposes the motion of a turbine is so regular that no automatic control is required, but for some classes of machinery a self-acting governor is desirable. The accompanying figure shows the one made by Messrs. Günther & Sons, of Oldham, for this purpose. It consists of an ordinary Watt governor (see Volume V.) which can shift a belt from a loose pulley to either of two fast pulleys connected by bevel gearing to the pillar for adjusting the turbine. At the normal speed the belt is on the loose pulley, but any change of speed causes the belt to be shifted and the vertical shaft is turned more or less in one way or the other according to the required quantity of water. The governor is driven from a belt on the turbine, or by some shaft from it, and can be disconnected from the hand gear by merely freeing a clutch.



GÜNTHER'S TURBINE GOVERNOR. *

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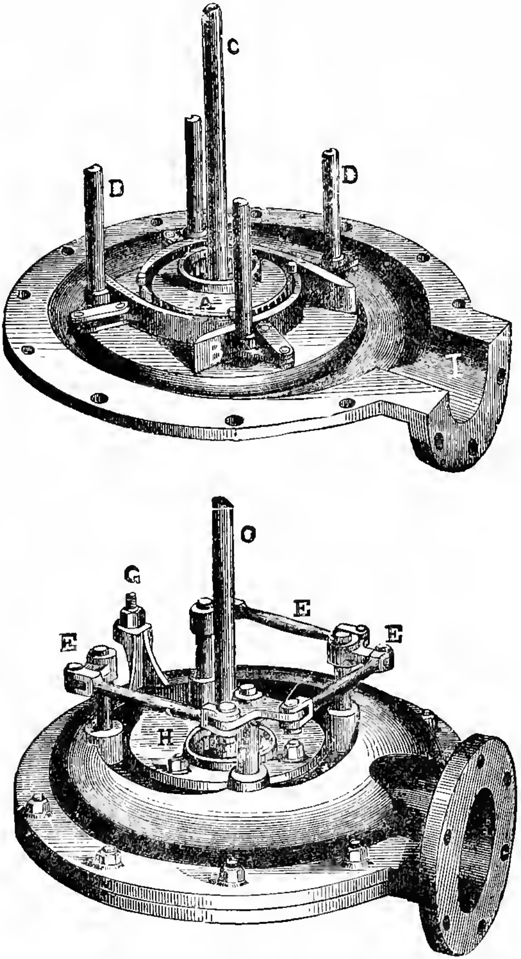
* See Notes at end of this Lecture.

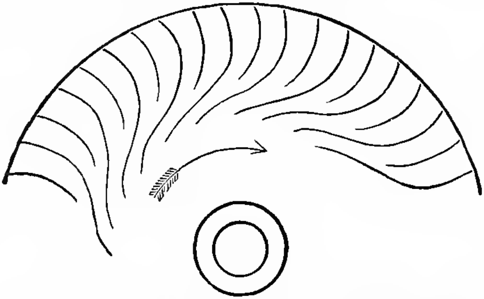
Thomson's Vortex Turbine.—Imagine a wheel placed with its axis coinciding with that of the free vortex already considered in the preceding lecture. This wheel will be carried round by the vortex, and if we arrange matters so that the water passes through the



Vortex Turbine by Messrs. Gilbert Gilkes & Co., Ltd., Kendal.

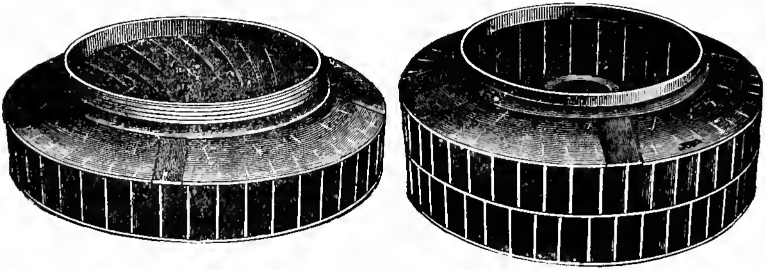
wheel giving up its energy to it while fresh water takes its place, we have the essentials of Professor James Thomson's vortex turbine.

As will be seen from the foregoing illustrations, there are only four guides in the vortex, and by altering the inclination of these, we can adjust the radial component of the motion of the water and the amount of water flowing through the turbine. In the first figure the turbine is represented with its cover removed, and in the second with its case complete. A is the revolving wheel keyed to the shaft C. B is one of the guide blades connected by the bell cranks and shafts D to the outside rods E, which can be adjusted by a screw or by a governor. The shaft runs on a lignum vitæ pivot which is lubricated by the water.



SECTION OF WHEEL OF VORTEX TURBINE.

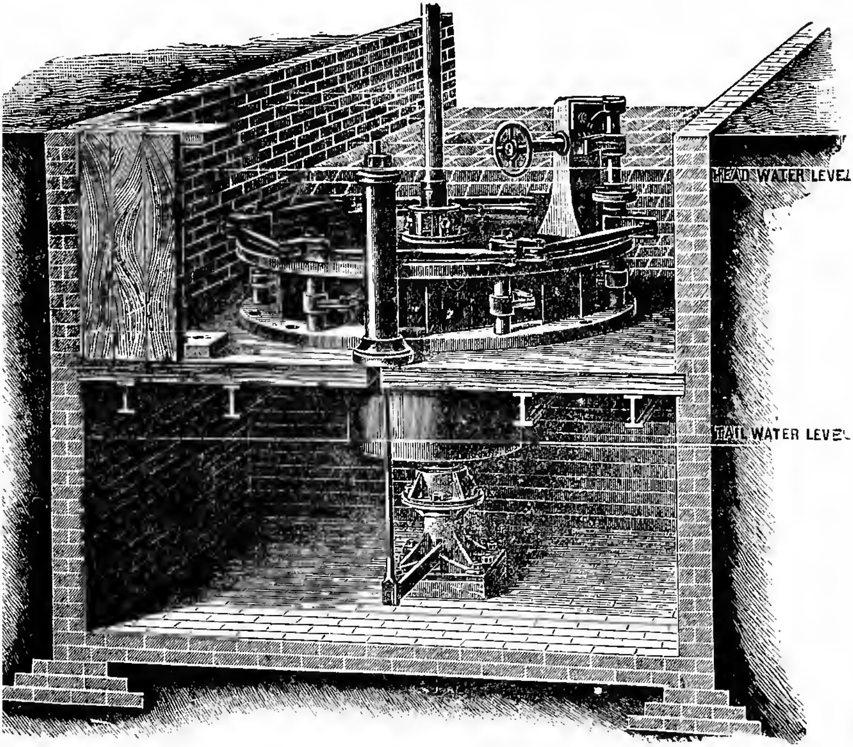
On account of the constrained motion of the water inside the wheel it requires a large number of guides. For the purpose of reducing the friction and to lessen the loss of area due to the thickness of the guides, it will be seen from the above section that every second guide is only half the length of its neighbouring one. The wheel is made either single or double. In the latter case, it



SINGLE AND DOUBLE VORTEX WHEELS.

has the same efficiency at half gate as at full gate, but being of the inward flow "drowned" type it will have a lower efficiency at other loads. Our next figure illustrates one of these turbines as fixed in position.

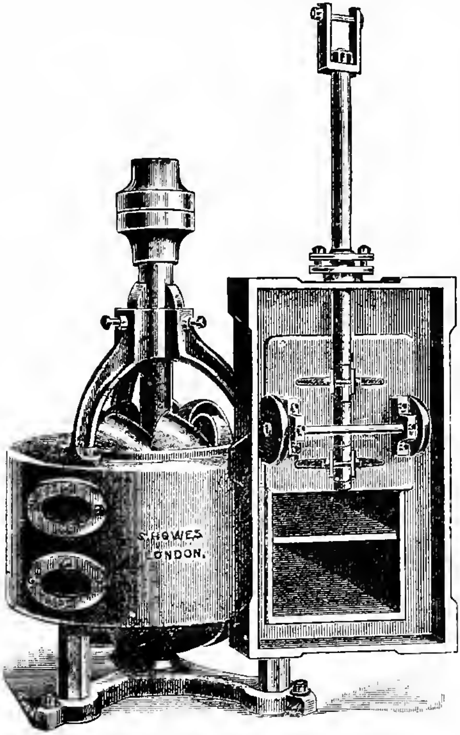
One great advantage of an inward flow turbine is that, to a certain extent, it is self-governing. When its velocity increases, so also does the centrifugal force which opposes and consequently reduces the inflow of the water; whilst the opposite action takes place when the turbine is being reduced below its normal speed.



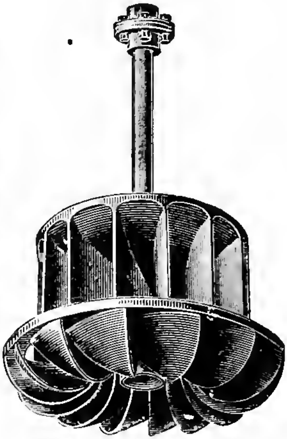
Single Vortex Turbine by Gilbert Gilkes & Co., Ltd , Kendal.

Little Giant Turbine.—We now come to the fourth or mixed flow type of turbine, and as an example we have chosen the "Little Giant" turbine. As may be seen from the illustrations the water enters at the circumference and passes out at the top and bottom and there is a sluice for regulating the supply. The passage has a division so that water can be entirely shut off from the upper half of the wheel.

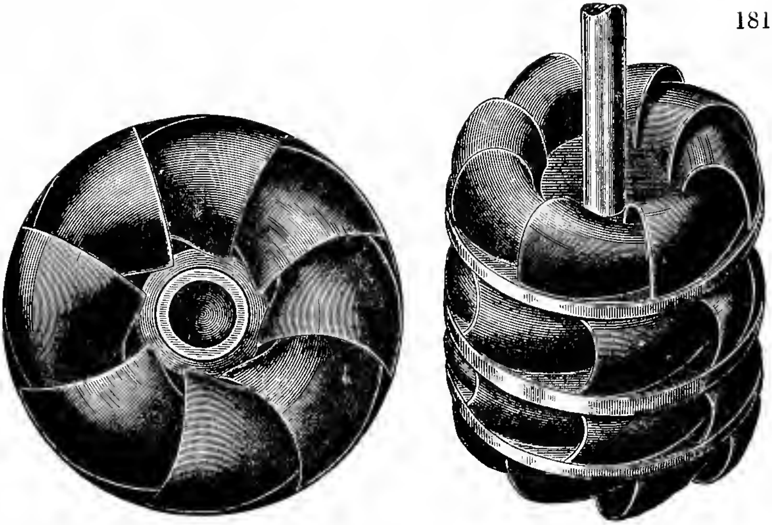
The same firm also makes a special "flume" turbine—*i.e.*, one which is placed directly in the water in the same way as the single vortex turbine shown on the previous page. The sectional view shows a turbine formerly made with a wheel similar to that in the Little Giant Flume Turbine, but with a different casing. In this instance, the pivot for the wheel can be raised or lowered by a lever which is clamped in position by a nut. The wheel is held down by a fixed ring fitting in a groove on the wheel, just above the lower splayed-out part.



Little Giant Turbine
by S. Howes, London.

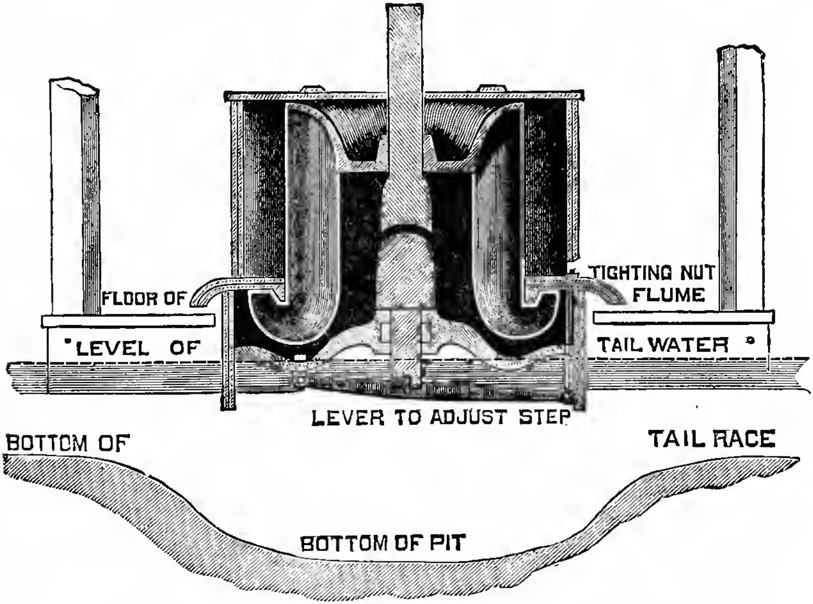


Wheel for Little Giant
Special Flume Turbine.



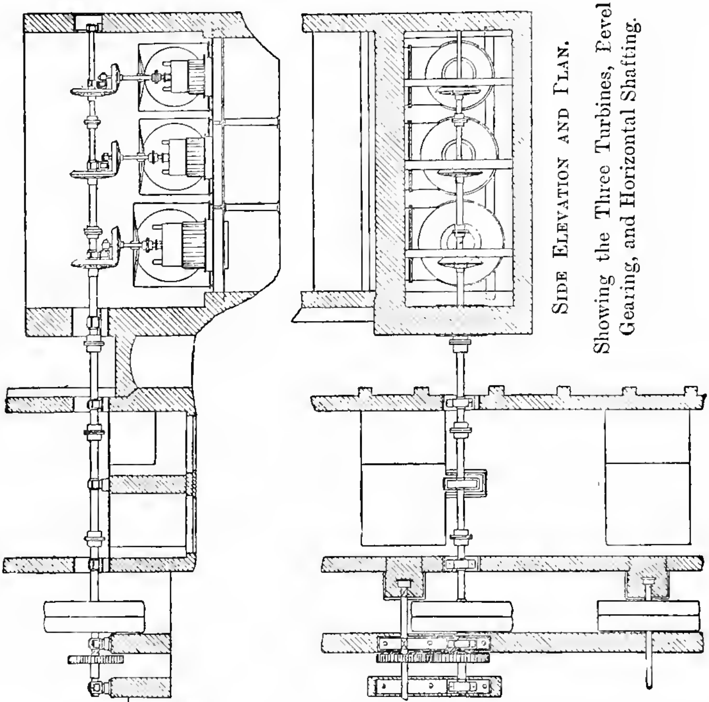
End View. Side View.

Wheel for Little Giant Turbine.



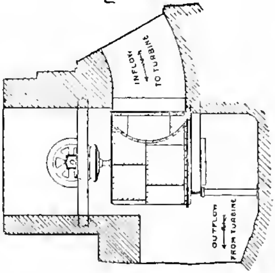
Vertical Section of the Little Giant Turbine.

Stoneywood Paper Works, Turbine Installation.*—This installation consists of three "Hercules" turbines, 54 inches, 48 inches,



Side Elevation and Plan.

Showing the Three Turbines, Bevel Gearing, and Horizontal Shafting.



Vertical Section through Outflow and Inflow.

Stoneywood Paper Works.

"Hercules" Turbine Installation of 1,200 H.P.

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* I am indebted to John Turnbull, Jun. & Sons, Glasgow, who put down and set agoing this plant, for the drawings from which the accompanying figures were made.—A.J.

and 42 inches diameter respectively. They are all worked by a fall of 23 feet, and the combined 1,200 H.P. of the three turbines is transmitted by bevel gearing to a horizontal shaft, which revolves at 83 revolutions per minute, as shown by the accompanying views. The power is then conveyed to the various departments of the paper works from this horizontal shaft, principally by belts, but in some cases by gear wheels.

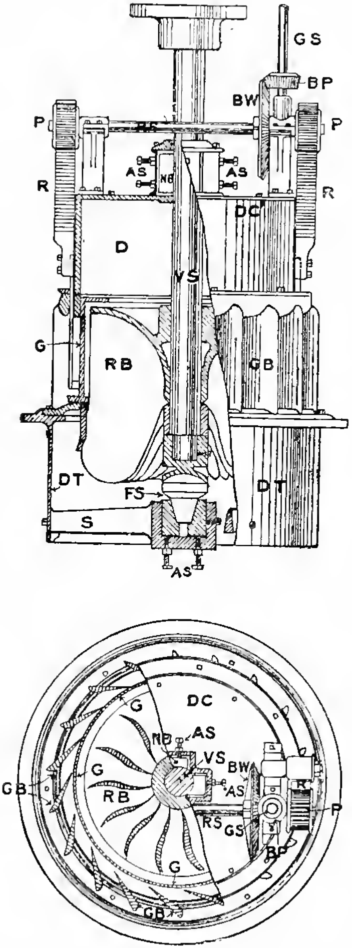
The water is supplied to the turbines through a head race, from the Aberdeenshire river, Don. The sectional area of this race is 22 feet wide by 5 feet 6 inches deep. After passing through the turbines, the water is discharged into a tail race which connects with the river at a lower level.

Hercules Turbine.—This low-pressure mixed flow turbine is of American origin and make. Many of these turbines are to be found doing good work in this country under different water heads up to 40 feet. They are made in twenty different graduated sizes, varying from 1 to 2,800 horse, of which the following table shows the capabilities of the former and the latter under the minimum and maximum pressures :—

	Head in Feet.	Horse-Power	Cubic Feet of Water discharged per Minute.	Revolutions per Minute.
No. 1 Size, . .	4 40	1·2 35	190 580	266 840
No. 20 Size, . .	4 40	90 2,830	14,800 46,800	38 117

From the previous and accompanying figures of this turbine, with index to parts, the construction and action will be easily understood. When the gate shaft, G S, is turned in the direction for lifting the gate, it does so through the bevel gear, B P and B W, pinions, P, and racks, R, which are bolted to the upper end of the thin steel cylindrical gate, G. This action permits the water from the source of supply to flow inwards upon the revolving buckets, R B, through the fixed guide blades, G B. Having given up the greater part of its energy to the vertical shaft, V S, the water discharges freely from the draft tube, D T, into and below the surface of the tail race, as shown by the left-hand arrow in the previous figures.

The makers of this turbine and their representatives maintain, that turbines of the "Vortex," and "Little Giant" types, previously described in this Lecture, cannot possibly give such a high efficiency at part gate as those with fixed guide blades. They also state, that turbines constructed with gates which, when partly closed, alter the angle of the inflowing water as it strikes or presses upon the buckets of the revolver or rotor, are less efficient than the mixed-flow type for the reason, that the angle of direction of the moving water in the latter is not altered at any part of the gate opening.



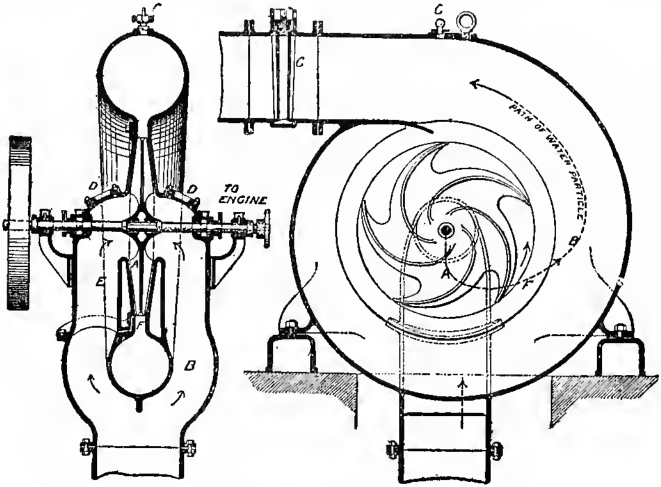
Improved "Hercules" Turbine.

Index to Parts.

G S for Gate Shaft.
B P ,, Bevel Pinion.
B W ,, Bevel Wheel.
R S ,, Rack Shaft.
P ,, Pinions.
R ,, Racks.
G ,, Gate.
G B ,, Guide Blades.
R B ,, Revolving Buckets.
V S ,, Vertical Shaft.
F S ,, Footstep.
S ,, Support for Footstep.
A S ,, Adjusting Screws.
D T ,, Draft Tube.
D ,, Dome.
N B ,, Neck Bush.

Further, that those turbines which are fitted with the cylindrical gate, as herewith illustrated, maintain a higher efficiency at all loads under the maximum power of the turbine than other forms of low-pressure turbines. They guarantee an efficiency of 80 per cent. from half to full gate, and with the larger sizes, they claim 85 per cent. efficiency for ratio of B.H.P. to E.H.P. due to waterhead.

Centrifugal Pumps and Fans.—If certain kinds of turbines be driven by a prime motor the centrifugal force of the fluid carried round with the wheel will cause the fluid to flow outwards in a continuous stream. If the fluid be water the machine is termed a *centrifugal pump;* whereas, if we are dealing with air, it is called a *fan.* Such machines are, therefore, merely reversed turbines; but, in order to get the best results, they must be specially designed, because the best design for a turbine is not by any means the most efficient and suitable for a pump or fan.



CENTRIFUGAL PUMP.

Centrifugal pumps are largely used whenever a quantity of water has to be elevated through a small height, such as in the case of emptying graving docks and sunken vessels, circulating the cooling water through condensers, or dredging sand, gravel, and mud from rivers, whilst centrifrugal fans are employed for ventilating mines, ships, and buildings, as well as for producing an artificial draught to smiths' fires, cupolas, and boiler furnaces, &c.

The illustration* shows a good form of centrifugal pump with curved blades. A is a wheel rotating inside a casing B, thus giving the water which enters at the centre a certain amount of kinetic and pressure energy. F is a small whirlpool chamber which, as already explained, allows the water to form a free vortex, and converts part of its kinetic energy into energy of pressure. The water moves in a free vortex in the volute-shaped pipe, the path of a partical being shown in the figure by a dotted line and arrows.

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* The above figure is reduced from one in Mr. Lineham's *Text-book on Mechanical Engineering* (Chapman Hall & Co.), by kind permission.

The difference of pressure produced by the pump depends upon the density of the fluid in the wheel as well as on the speed of rotation. Consequently, when there is only air in it, the pump is not able to produce a sufficient vacuum to make the water rise into it. In order to get over this difficulty, an ejector G, and a sluice C, are added. When the pump is to be started the sluice is closed and the air exhausted from the pump chamber by a jet of steam being passed through the ejector. The water then rises into the wheel and the sluice is gradually opened as the speed is increased. When the pump is fairly started the steam jet is shut off and the sluice fully opened. Sometimes a non-return valve is placed at the foot of the suction pipe to prevent the pump and pipe emptying when the wheel is stopped. In such a case, the pump is ready to start again at once.

Sometimes centrifugal pumps are made with radial blades. They then require a much larger whirlpool chamber to allow the kinetic energy to change into pressure energy without a serious loss in eddies.

Note 3 on the Second Edition, by Mr. J. Ritchie, Waverley Engineering Works.

Pelton Wheels.—"Your description of the action of a jet on the Pelton bucket is very clear, as it illustrates the flow of the water over the bucket in a way which I have never met with before. It does not, however, in my opinion represent the true direction of the water along the face of the bucket. It would be all right if the buckets were fixed to a chain passing over two pulleys, where they would move away in a straight line from the jet, until completely emptied. They move, however, in a circle, and the centrifugal force brings into action another movement in an outward direction to the point of the bucket, and part of the water is, therefore, discharged over the lip of the bucket, as well as at the sides.*

"The accompanying figure shows a small Pelton wheel bucket, from which I have got very good results. With a 12-inch wheel at 1,350 revolutions per minute and a head 310 feet, it gives 8 horse-power.

"I have never been able to get the very high efficiencies stated by the Pelton Wheel Company. Even with polished bronze wheel buckets, 75 to 80 per cent. is the highest which I have obtained.

Pelton Wheel Regulation.†—"I think something might be added on the subject of regulation. The weak point, of course, is, as you point

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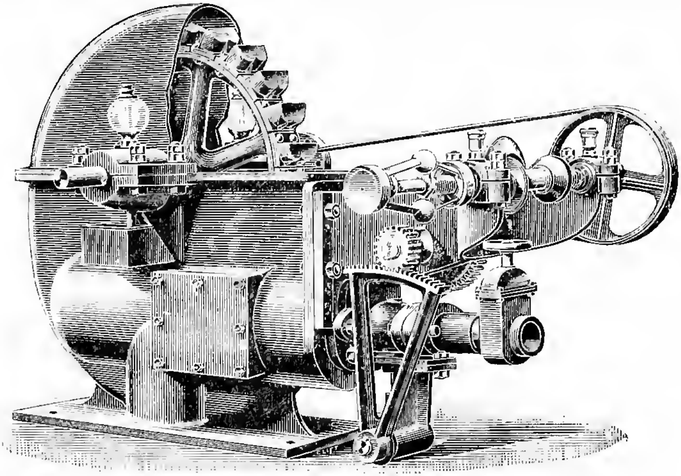
*True; but the three positions of the Vanes in my illustration are supposed to be *very close* to each other.—A.J.

† Students may here refer to a paper on "Experimental and Analytical Results of a Series of Tests with a Pelton Wheel," by William Campbell Houston, B.Sc. See vol. xlvi., part viii., of *Transactions of the Institution of Engineers and Shipbuilders in Scotland*, Glasgow, April, 1903.—A.J.

out, the waste of water by throttling. The nozzle has to be made large enough to do the maximum work, and unfortunately at the same time that the greatest quantity of water is passing, the net or true head is generally the least, since the frictional losses in the pipe are then greatest.

"I have adopted a central spear to reduce the area of the jet and find it to be one of the most efficient methods, since the jet becomes thereby partly annular. The accompanying figure shows the type of this wheel fitted with a regulating nozzle under the control of a governor.



Pelton Wheel with a Governor.
(By Carrick & Ritchie, Edinburgh.)

"One of the principal losses with Pelton wheels is due to the bucket striking the jet, whereby a portion of the water is momentarily diverted as the bucket is entering the jet. Many different forms of buckets have been devised to avoid this loss, but without much success.

"In double or treble nozzle wheels it is advisable to allow at least one bucket to intervene between the two that may be under the action of the jets. This is to allow of the action which you describe on p. 507, where the water continues to travel over the bucket, when the jet is acting on the following vane.

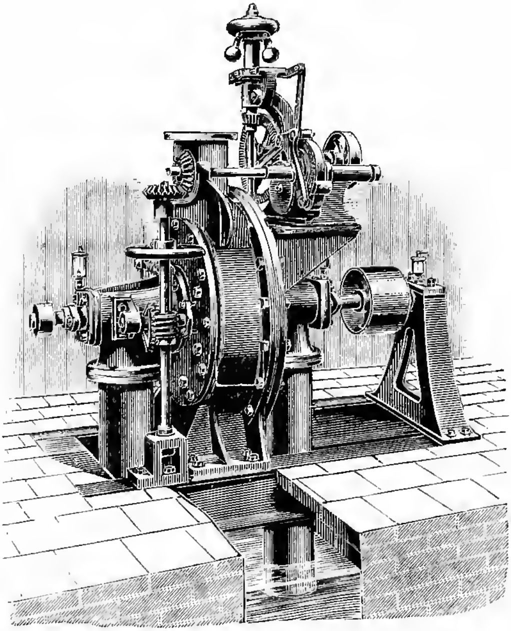
"The use of Pelton wheels is rapidly extending for high falls. The efficiency with a good regulating nozzle is as high as a Girard turbine. They are less complicated and cheaper.

Girard Turbines.—"The description and illustrations are all good. It may be mentioned that outward-flow Girard turbines are very difficult to govern, owing to the centrifugal force acting on the water passing through the wheel, increasing the flow instead of diminishing it, as in the inward-flow turbine.

"I have made a great many of the outward-flow Girard type of turbine with horizontal shafts. I have not found the maximum efficiency to be so high as in the case of the inward-flow or pressure and reaction type of turbines. It is not advisable to use the latter for heads over 200 feet: whereas, I have Girard turbines working under 800 feet. At the same time there is no doubt, that for great variations—say down to one-fourth of the supply—there is nothing to beat the Girard turbine, owing to the method of regulation—viz., shutting off the supply from each port, one after the other.

Jonval Turbines.—"I notice that on the Continent the Jonval turbine is being rapidly superseded by mixed-flow turbines. The speed of the former is necessarily slow, owing to its comparatively large diameter, which naturally involves heavier gearing and larger pits, &c., than with the latter.

Turbine Governors.—"The governor shown at pp. 172, 176 is a good one, but it is not quite sensitive enough for electric lighting work. In order to move the belt, the governor itself must be slow and heavy. King's patent governor, which is shown on the high-pressure double-discharge Waverley turbine, is a very sensitive high-speed governor. All the work this governor has to do, is to move the cover plate of the pawl wheel.



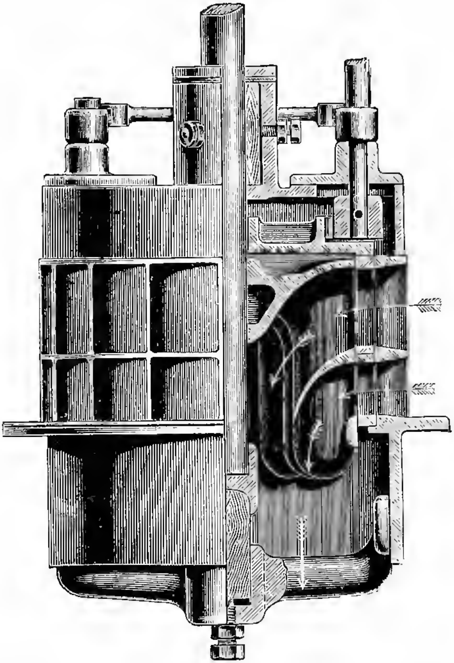
High-Pressure Double-Discharge Waverley Turbine with King's Governor.

"Owing to water being a non-elastic body, turbines and Pelton wheels are much more difficult to govern than a steam engine. The principal difficulty to contend with is hunting, as the action of the governor, owing to the momentum of the parts, is continued after the reason for the action ceases, and the governor, therefore, keeps turning 'off' or 'on' the water longer than is necessary. The relay arrangement is calculated to prevent this.

Inward-Flow Turbines.—"Thomson's Vortex turbine is the parent of all the inward and mixed-flow turbines in use at the present day. It has a very good efficiency. The principal objection to it is the choking which takes place at the centre of the wheel. Consequently, it has to be made of large diameter and comparatively shallow, compared with other turbines of equal power; but for many purposes it still holds its own.

Mixed-Flow Turbines.—"I must frankly say I do not think you have been fortunate in selecting for illustration the best type of a mixed-flow turbine. Owing to the attempt which most American manufacturers have made, to get the greatest possible quantity of water through the wheel for a given diameter, the designers of this wheel have introduced such sharp curves, that the frictional losses are great. There are no guides in the proper sense of the term, the casing being in the form of a scroll. The water assumes a circular or semi-vortex motion in the same direction, before entering the buckets of the wheel.

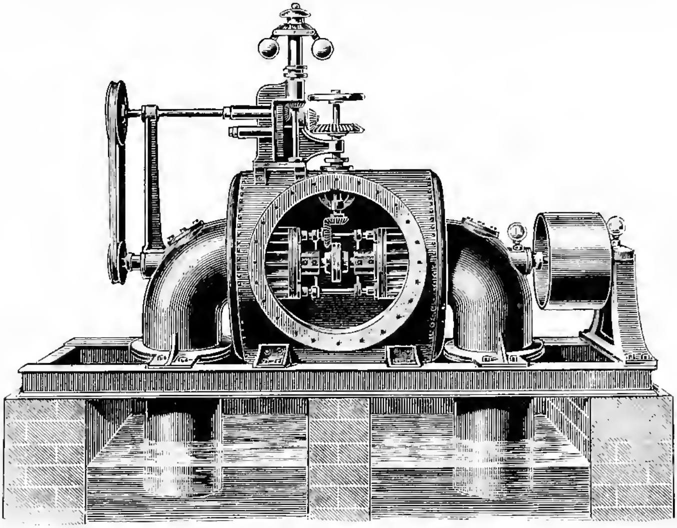
Waverley Mixed-Flow Turbine.—"The usual form of this turbine with a vertical shaft, suitable for fixing in an open flume is shown by the following figure. The special feature of the Waverley turbine is the



Waverley Mixed-Flow Turbine with Double Bucket.
(By Carrick & Ritchie.)

division in the buckets. This division may be placed in any position from one-third to one-half down the bucket, and is specially of advantage where the water supply is variable. The usual form of gate is cylindrical and it is placed between the wheel and the guide ring. The upper portion of the wheel may be entirely shut off in the event of the water being low, and the efficiency of the wheel is thereby increased. In the Waverley wheel about one-half of the water is discharged radially and the other half axially, through the lower part of the buckets.

"The cylindrical gate of the turbine is operated by a pinion and rack.* The accompanying figure shows a duplex turbine, consisting of two wheels on one shaft. This arrangement has the advantage of doing away with the end thrust on the shaft as the pressure on the one wheel is balanced by the other.



HORIZONTAL DUPLEX WAVERLEY TURBINE.

"This type of turbine is suitable for falls up to 50 feet, but the former figure with King's governor shows a turbine of a somewhat similar construction with a double-discharge pipe, which is built of sufficient strength for falls up to 200 feet. These turbines are very suitable for the direct driving of dynamos. For example, in connection with the electric lighting of Blebo House, near Dairsie, Fifeshire, I used one where the head is only 26 feet, and 10 feet of this head is due to the 'draft or suction pipe.' It has two gun-metal wheels 9 inches diameter, and gives 15 H.P. at 680 revolutions per minute. The lamps are run direct from the dynamo, for there are no accumulators, and the speed of this turbine is entirely controlled by a King's governor.

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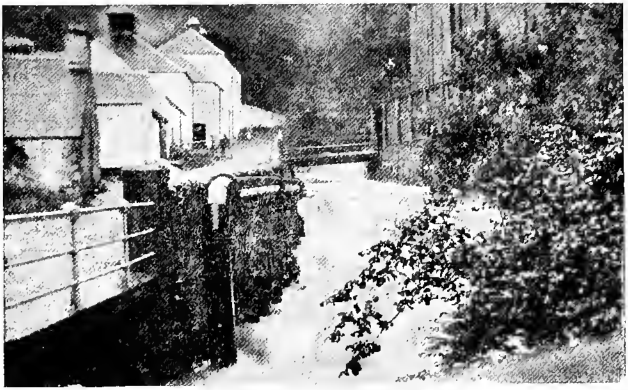
* This is a similar arrangement to that already described in this Lecture in connection with the "Hercules" turbine.—A.J.

"The forms of the guides and buckets of mixed-flow turbines are not the result of any elaborate 'scientific' calculations, but are based on the general assumption, that the greatest power is got out of the water by turning it as nearly in an opposite direction to its flow, or through 180°. Of course, mechanical difficulties make the latter impossible, but the mean angle through which the water is turned is about 125°. The body of the wheel being a cone, the water is at the same time given a vertical direction, the lower portion of the bucket being curved backwards. The mean angle of discharge is about 20°, the extreme end of the bucket is only about 16°, so that the water in this part of the wheel takes a downward, and also slightly outward direction.

"The best form of buckets and guides are simply the result of innumerable tests, in which the forms which gave the highest results have been retained.

"In mixed-flow turbines it is usual to have an unequal number of buckets to that of the guides. This, with the slight curve given to the edge of the bucket, is found to give a more regular turning moment, than if all the ports and buckets came exactly opposite each other at the same moment. One great advantage of the mixed-flow turbine is the ease with which a draft pipe may be used in connection with it. At Tobermory, Scotland, I put down a horizontal draft or suction pipe with a total length of 90 feet, in order to obtain a final vertical drop of 12 feet into the tail race of the river. The head due to this long draught pipe is nearly as efficient as if the same head were on the pressure side of the wheel."*



TURBINE INSTALLATION AT TOBERMORY.
(Showing Long Draft Pipe.)

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* See my Question 21 on this point at the end of this Lecture.—A.J.

LECTURE V.—QUESTIONS.

1. Sketch an undershot wheel. Explain why its efficiency is so low when it has radial blades, and show how the blades should be made to avoid this loss.

2. Water flows radially at 4 feet per second towards a part of a wheel of a centrifugal pump or turbine which is moving at 12 feet per second, find the angle of the vane that the water may enter without shock. If the vane were radial, at what angle ought the water to be guided so that it might enter without shock; its radial component of velocity being the same as before?

3. Give outline sketches of the common types of water-wheel, and compare their relative advantage.

4. Distinguish between water-wheels and turbines, and explain the advantages of the latter.

5. Sketch the Pelton wheel and describe its action.

6. Sketch and describe a turbine of the Girard type, and mention its advantages and disadvantages.

7. Describe, with sketches, a Jonval turbine, and explain its relative advantages.

8. Sketch the wheel and case of an inward flow turbine for a fall of 50 feet; 8 cubic feet of water per second. Calculate the diameters and breadths of the wheel, the number of revolutions per minute, and the size of the shaft.

9. The vane of a wheel of an internal flow turbine is normal to the rim which moves at 40 feet per second. Water is guided so as to enter the rim with a radial velocity of 4 feet per second; what must be the angle made by the guide blade with the rim, if the water is to enter without shock? If the circumferential area of all the openings of the rim is 1·5 square feet, what volume of water flows per second? We use convenient but not very exact language when we ask—what is the total tangential momentum per second entering the rim?

10. Sketch an inward flow turbine for a fall of 30 feet and 20 cubic feet of water per second. Give its principal dimensions and speed and the angles of guide blades and vanes What is the kinetic energy of a pound of water just entering the wheel? Neglect losses by friction.

11. The radial velocity of water in a centrifugal pump wheel is 2 feet per second. The vane makes an angle of 35° with the circumference; what is the velocity of the water *relatively to the wheel?* The wheel is 2 feet external diameter, and makes 300 revolutions per minute; what is the actual velocity of the water as it leaves the wheel? What is the circumferential momentum of each pound of water? Neglecting radial velocity of water, what is the work done to every pound of water if it enters the wheel with only a radial velocity. and what is the kinetic energy of every pound of water? *Ans.* 3·48 feet per second; 28·63 feet per second; 0·89 lb.-ft.; 27·9 ft.-lbs. (B. of E. Adv. & H., Part I., 1900.)

12. Ten cubic feet of water per second enters a turbine wheel with a tangential velocity of 50 feet per second; it enters without shock, the velocity of the rim of the wheel being 50 feet per second; the water leaves the wheel with only a radial velocity. What energy does the water give to the wheel per second? (B. of E. Adv., 1901.)

13. Water, with a radial velocity of 6 feet per second, is really moving along a vane which makes 30° with the circumference of a centrifugal pump wheel, which lets the water leave the wheel with less tangential velocity than that of the wheel itself. Show on a diagram and state the tangential velocity of the water if that of the wheel is 40 feet per second. If the water before it enters the wheel has a radial velocity of 6 feet per second, and enters without shock, what energy is gained per pound of water? What is the gain of kinetic energy? What difference will friction of the passages make in your result? (B. of E. H., Part I., 1901.)

14. Deduce an expression for the efficiency of a simple hydraulic reaction wheel when you are given the available head and the velocity of the water at the discharging orifice. What will the efficiency be if the head equals 112 feet and the velocity of discharge is 90 feet a second, neglecting friction? (C. & G., 1900, H., Sec. C.)

15. Describe, with sketches, a Pelton waterwheel. In what circumstances is such a waterwheel used? What is likely to be the efficiency of the wheel in ordinary working? (C. & G., 1901, H., Sec. C.)

16. A Pelton wheel, 2 feet in diameter, runs at 600 revolutions a minute, the available pressure at the nozzle being 200 lbs. per square inch, and the available supply 100 cubic feet per minute. Estimate the available horse-power and the greatest possible hydraulic efficiency of the wheel. (C. & G., 1902, H., Sec. C.)

17. Discuss, in general terms, the various losses in a centrifugal pump, pointing out in particular the effect of the whirlpool chamber and of the angle of discharge of the vanes on the efficiency. In a test of a compound centrifugal pump the pressure in the rising main, measured at the pump, was found to be 55·8 lbs. per square inch above atmosphere, and on the suction pipe, measured at the pump, 6·7 lbs. per square inch below atmosphere. The water raised per minute was 119 gallons, whilst the turning moment transmitted to the pump spindle, as measured by a dynamometer, was found to be 27·3 foot-lbs., the revolutions being 1460 per minute. Find the lift, the horse-power of the pump, and the efficiency. (C. & G., 1902, H., Sec. C.)

18. Explain what is meant, in referring to hydraulic pressure engines, by the statement that "they contain within themselves their own brakes." It is found that a ram of 10 inches in diameter can lift a load of 20 tons with an uniform velocity of 6 inches per second when supplied with water from an accumulator working at a pressure of 750 lbs. per square inch. Find the coefficient of hydraulic resistance referred to the velocity of the ram, and, assuming this to remain constant at all speeds, find the velocity of the ram when under no load. (C. & G., 1902, H., Sec. C.) (*For answer to first part refer to the Froude Water Dynamometer or Brake described in Vol. I.*)

19. Explain, with the aid of sketches, the various devices employed to make water-pressure engines economical when working under loads less than their maximum. (C. & G., 1902, H., Sec. C.)

20. The following results were obtained from an undershot water-wheel of 15 feet diameter, 4 feet 6 inches wide, and having only a head of water of 3 feet 6 inches:—

Revs. per Minute.	B.H.P.	Revs. per Minute.	B.H.P.
7·3	6·68	8·9	6·68
8·3	6·7	9·2	6·48
8·4	6·7	10·0	6·4
8·6	6·73	...	...

Plot the circumferential speeds, and the corresponding B.H.P. as co-ordinates. Then, find a constant for the ratio of the velocity of rim of wheel to its diameter at the most effective speed in this case. Also, find the ratio of the speed of rim of wheel to the velocity of the water due to the above head. Can you refer to any reliable data regarding these two ratios or other applicable ratios for water-wheels?

21. Given two turbines with their feed-water pipes identically the same in every respect. Let each turbine be supplied with water from, say, a height of 20 feet above their respective centres. In case (1), the turbine discharges directly into the air; whereas, in case (2) the bottom end of the discharge pipe is led into the tail race below its surface, and the vertical distance from this surface to the centre of the turbine is 20 feet. Which will yield most power, why, and by how much? Sketch and explain your answer. If the centre of the turbine in case (2) were 30 or more feet above the surface of its tail, what would happen, and why?

N.B.—*See Appendices B and C for other questions and answers.*

LECTURE V.—A.M.INST.C.E. EXAM. QUESTIONS.

1. What are the two conditions to be observed in designing the angles of the vanes at the inlet and outlet surfaces of a turbine to secure the greatest efficiency? Define an impulse and a pressure turbine. (I.C.E., *Oct.*, 1897.)

2. Describe the construction of any one form of turbine, stating how the power and consumption of water is regulated. Discuss the effect of regulating the flow through the turbine on the efficiency. (I.C.E., *Oct.*, 1897.)

3. Show that the hydraulic efficiency of a turbine which works on a fall of H feet is $\eta = w_1 \mathrm{V}_1/g\,\mathrm{H}$, where w_1 is the velocity of whirl, and V_1 the velocity of the wheel at the receiving surface, the water being rejected radially. (I.C.E., *Oct.*, 1897.)

4. In an inward flow turbine the water enters the inlet circumference, 2 feet diameter, at 60 feet per second, and at 10° to the tangent to the circumference. The velocity of flow through the wheel is 5 feet per second. The water leaves the inner circumference, 1 foot diameter, with a radial velocity of 5 feet per second. The peripheral velocity of the inlet surface of the wheel is 50 feet per second. Find the angles of the vanes at the inlet and outlet surface. (I.C.E., *Feb.*, 1898.)

5. Describe some different forms of turbine, and mention any special advantages of each arrangement, or any particular conditions for which each is most suitable. Describe specially the regulating arrangement of each turbine. (I.C.E., *Feb.*, 1898.)

6. Describe any one form of turbine governor. State what are the special difficulties in applying governors to turbines, and point out how the difficulties are met in the governor you describe. (I.C.E., *Feb.*, 1898.)

7. Describe how you would measure the actual power given out by a water-motor, or any other motor, and the power supplied to it, and so arrive at its efficiency. (It may be well to say that if the Candidate has not actually measured the brake-power of a machine, he will be apt to make a mistake in the description.) (I.C.E., *Oct.*, 1898.)

8. A horizontal jet of water impinges on a vane rotating about a horizontal axis above it, when the arm of the vane makes an angle of 120° with the jet, sketch the shape of vane to obtain the maximum amount of energy out of the jet, and indicate the path of the water on leaving the vane. (I.C.E., *Oct.*, 1898.)

9. Deduce the general formula for turbine efficiency. (I.C.E., *Oct.*, 1898.)

10. A vane, circular in section, subtending an angle of 30° at its centre, rotates about that point, in a vertical plane, with a linear velocity of 3 feet per second. A jet of water, 12 square inches in section, impinges upon it tangentially with a velocity of 9 feet per second. Find the power transmitted by the jet to the vane. (I.C.E., *Feb.*, 1899.)

11. Water under a head of 100 feet flows at the rate of 10 feet per second through a horizontal bend-pipe 36 inches diameter and 20 feet radius, subtending an angle of 60° at the centre of the curve; calculate the total pressure tending to force the bend-pipes outward. (I.C.E., *Feb.*, 1899.)

12. Describe the construction of an impulse turbine, and state how it is regulated for supplies below the normal. (I.C.E., *Feb.*, 1899.)

13. Sketch a centrifugal pump and state what you know of the conditions upon which its efficiency depends. (I.C.E., *Oct.*, 1899.)

14. It is desired to utilise a stream of water having a head of 40 feet, measured from the rock over which it falls to the level of the stream below. The flow is exceedingly variable, and while it is desired to make the most of the minimum flow, amounting to about 5 cubic feet per second, advantage must be taken of at least four times that flow when it occurs. (*a*) Between water-wheels and turbines, and their different classes, how would you choose, and why? (*b*) What brake horse-power should you expect to obtain from the minimum flow? (I.C.E., *Feb.*, 1900.)

15. Explain, with as few mathematical symbols as possible, the fundamental conditions of mechanical efficiency in turbine action, taking account both of the waste pressure and kinetic energy in the water-discharge and of the frictional wastes. (I.C.E., *Oct.*, 1900.)

16. What are the latest developments in the use of centrifugal pumps for force-pumping against a large head? Describe briefly the dynamics of this use of centrifugal pumps, and more particularly in respect of the mode of generation and accumulation of high pressure and in respect of mechanical efficiency. (I.C.E., *Feb.*, 1901.)

17. Sketch good forms of admission and exhaust valves for hydraulic pressure motor engines. (I.C.E., *Feb.*, 1901.)

18. Explain fully how the energy of water is utilised in an over-shot water-wheel, in a Pelton wheel, and in a reaction turbine. (I.C.E., *Oct.*, 1901.)

19. Indicate how a vane, moving with a velocity of 25 feet per second in a horizontal direction, must be shaped in order to abstract the maximum amount of energy from a jet of water impinging upon it at an angle of 45° to the horizontal with double the above velocity. What pressure would be exerted on the vane per cubic foot of water impinging per second? (I.C.E., *Oct.*, 1901.)

20. Explain the action of a Girard type of impulse turbine, and indicate how with a vertical shaft the pivot supporting the weight of the turbine wheel can be made easily accessible for lubrication and adjustment. (I.C.E., *Feb.*, 1902.)

21. In an outward flow impulse turbine the jet of water, moving with a velocity of 150 feet per second, is guided on the wheel vanes at an angle of 75° to the radius, the inside radius of the wheel being 5 feet and the outside radius 6 feet. If the wheel makes 150 revolutions per minute, and the water leaves the wheel with a radial velocity only, draw the shape of the vanes. (I.C.E., *Feb.*, 1902.)

22. Show how the hydraulic efficiency of a turbine may be found from the principle that the angular impulse is equal to the change of angular momentum. (I.C.E., *Feb.*, 1902.)

23. Calculate the maximum possible efficiency of a Pelton wheel in which the bucket velocity is three-quarters of the jet velocity. How much less than this would the actual efficiency be in the case of a wheel of about 40 H.P.? To what, and in what proportion, would you debit the losses? (I.C.E., *Oct.*, 1902.)

24. Explain briefly the apparatus you would use, and the calculations you would make, to determine the net efficiency of a turbine giving about 20 B.H.P., the shaft being vertical. (I.C.E., *Feb.*, 1903.)

25. For an inward-flow turbine, given the angle of the guide-blades, the speed of the vanes, and the radial velocity of the water at both inlet and

outlet, show how you would find the angles of the vanes at inlet and outlet, and how you would proceed to design them. (I.C.E., *Feb.*, 1903.)

26. Obtain a general expression for the hydraulic efficiency of a turbine in terms of the head, the angular velocity of the vanes, the tangential components of the velocity at inlet and outlet, and the radii of the inlet and outlet surfaces. Write down the corresponding formula for a centrifugal pump. (I.C.E., *Feb.*, 1903.)

N.B.—*See Appendices B and C for other questions and answers.*

LECTURE VI.

REFRIGERATING MACHINERY.

CONTENTS.—Refrigeration—Preliminary Considerations—Carbon Dioxide as a Refrigerating Agent—Elementary Refrigerating Apparatus—Simple Refrigerating Machine—Carbon Dioxide Refrigerating Plant—Anhydrous Ammonia as a Refrigerating Agent—De La Vergne's Refrigerating Plant—De La Vergne's Double Acting Compressor—The Linde System of Refrigeration—Apparatus for Transmitting the cold produced to the Chambers requiring Refrigeration—Pneumatic Tools—Questions.

Refrigeration—Preliminary Considerations.—An interesting example of the conversion of heat into work is afforded by a refrigerating machine. The simplest form of machine consists of an air-compression pump driven by a steam engine, or other motive power, in which the pump is water jacketted and the air is cooled under pressure by being passed through a surface condenser where the water abstracts the sensible heat generated by the mechanical work of compression. The air thus cooled, but still under pressure, is conveyed to an air engine and allowed to perform work by expanding against some resistance. A large proportion of the work originally performed during the operation of compression is again given out, with a corresponding reduction in the air temperature. A machine on this principle may be conveniently constructed by arranging the steam, compression and expanding engines to work on one crank-shaft. The expanding air thus assists in the work of compression. After deducting the necessary losses due to cooling, leakage, &c., the work done in the expansion cylinder amounts to about 65 per cent. of the power absorbed by the compression cylinder; the remaining 35 per cent. being supplied by the steam or other prime mover. The air having thus given up its heat, exhausts from the expansion cylinder at a very low temperature, reaching in one authenticated instance as low as $-124°$ F.

The large coal consumption of machines of this class has, however, led to their being superseded, in almost all cases, by machines in which a more direct conversion of heat into work takes place.

If we take any liquid and commence to vaporise it, we find that, it is necessary to maintain a continual application of heat in order to bring about this physical change. The amount of heat necessary to convert a unit weight of a liquid to a unit weight of gas at the same pressure, is always constant for the same liquid. For example, 1 lb. of water at a temperature of 212° F. and at atmospheric pressure, requires the application of 966·6 British Thermal Units to convert it into 1 lb. of steam at the same temperature and pressure. Conversely, to condense 1 lb. of steam to 1 lb. of water, both being at 212° F. and 14·7 lbs. pressure per square inch, we must abstract from the steam 966·6 thermal units by contact with a cold body. This principle holds good for any liquid.

A refrigerating machine with steam as a working medium, would not be practicable unless the temperature of everything in connection with it was maintained above 212° F., but there are many liquids which have, when compared with water, a very low boiling point; notably ether, sulphurous acid, ammonia, carbon dioxide, and ethylene. Each of these has been employed for purposes of refrigeration with more or less success; and all of them depend on the same principle—viz., the absorption or the giving out of their latent heat in converting the liquid to a gas, or *vice versâ*.

It is not necessary here, to enter into a discussion as to the relative merits of different liquids as refrigerating agents, but in practice, anhydrous ammonia is the agent generally used, and in a lesser degree, carbon dioxide. The necessary machinery is of itself extremely simple, although the details afford scope for a great amount of elaboration and ingenuity.

Carbon Dioxide as a Refrigerating Agent.—Carbon dioxide or carbonic anhydride, which is commercially known as "carbonic acid," is a colourless gas, and quite without odour when pure. It is under all circumstances perfectly innocuous, and has practically no effect on animal tissues or other bodies. It will, however, produce asphyxiation in animals when present in the atmosphere in quantities exceeding 25 per cent. by excluding oxygen from the blood. This gas may be very readily liquefied, either by diminishing its temperature or by increasing its pressure. This fluid has a specific gravity of about ·8, and can only remain in the liquid state when under considerable pressure, the pressure varying with the temperature of the liquid.* The moment the pressure is removed, the heat present in surrounding bodies, at once assists in the evaporation of the liquid carbon dioxide and the bodies themselves are consequently left in a colder condition than before the evaporation took place.

Elementary Refrigerating Apparatus.—Let us consider for a moment an elementary piece of apparatus in which refrigeration can take place. If we take two strong coils of piping and surround each with a vessel of water and then connect the two by means of a stop valve at their lower ends, as shown by the accompanying figure, we shall have a simple form of refrigerating machine. Suppose, that when the stop valve is closed, we charge the condenser coil with gas under liquefying pressure, by means of a force pump. The water surrounding the coil will absorb the heat which has been imparted to the gas by compression, and the condensed liquid will gradually accumulate at the bottom of the coil. On opening the stop valve, this liquid will run into the second or evaporating coil, and the pressure here being lower than is necessary for maintaining the liquid state of the material, evaporation will at once commence. The heat necessary for evaporating this liquid is absorbed from the water surrounding the evaporating coil, which will thereby become considerably reduced in temperature. To accomplish this result in practice three things are necessary :—(1) A compressor, to raise the pressure of the gas to whatever may be necessary for its liquefaction; (2) a surface condenser, to remove the heat generated by the mechanical work of compression; (3) an evaporating vessel, where the liquid may re-evaporate into a gas, and absorb heat in the operation.

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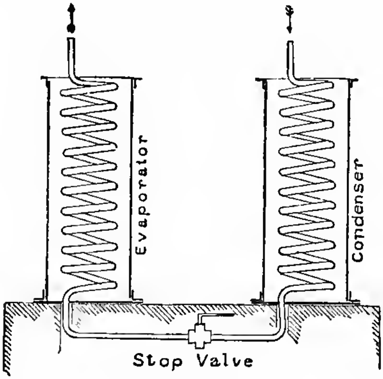
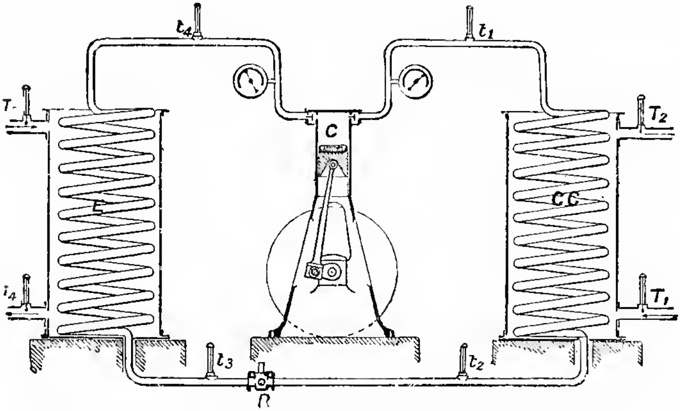
* Carbonic acid gas can only be liquefied by pressure when below 86° F., which is termed its critical temperature.



ELEMENTARY REFRIGERATING MACHINE.

Simple Refrigerating Machine.—The following figure of a simple refrigerating machine will explain the cycle of operations. C is a compressing pump delivering gas under pressure into the condensing coil C C, which consists of a strong worm of iron or copper piping immersed in a tank of water. R is a regulating-stop valve having a fine adjustment. E is the evaporator which consists of a coil of piping similar to the condensing coils. It is also immersed in a tank containing the water or other fluid to be cooled. The regulator R is closed, as soon as the pressure in the condenser has risen to that at which liquefaction can take place and the gas commences to condense on the inner surface of the coil C C. The drops of liquid descend and accumulate in the lower portion of this coil. The regulator is then opened, with the result, that a small quantity of liquid escapes into the evaporator. Now, since the compressor draws its supply of gas from the evaporator, the pressure in the evaporator must be less than in the condenser. Consequently, the liquid commences to boil, and absorbs heat for its transformation into a gas from the surrounding liquid. The temperature of this liquid is therefore naturally reduced by the operation. The liquid within the coil is entirely re-converted into a gas which ultimately finds its way to the compressor, and thus the cycle of operations is completed.



SECTIONAL DIAGRAM OF A SIMPLE REFRIGERATING MACHINE.

INDEX TO PARTS.

C for Compressor.
C C ,, Condensing Coils.
R ,, Regulating Valve.

E for Evaporator.
t_1 to t_4 ,, Thermometers.
T_1 to T_4 ,, Thermometers.

Suppose four thermometers be inserted into the pipes conveying the gas to and from the condenser and evaporator, as shown at t_1, t_2, t_3, t_4. It will be found that they do not register alike, for t_1 will show the highest temperature, then t_2 and t_3 will be some degrees lower, and t_4 will be lowest of all. Suppose now, that the liquids in the vessels surrounding C C and E be caused to circulate in the direction of the arrows, and that thermometers T_1, T_2, T_3, T_4, be placed on each of the inlet and outlet pipes. It will be found, that the temperature of the incoming water T_1 is lower than T_2 the temperature of the water going out of the condenser; also, that T_3 the temperature of the liquid entering the evaporator is *higher* than T_4 its temperature as it leaves this vessel. This shows, that with respect to the gas or liquid within the coils of the condenser or evaporator, heat is lost in the condenser and gained in the evaporator. The amount of the former is represented by the difference between T_1 and T_2 multiplied by the weight of water passed through the condenser in pounds, and the latter may be expressed in terms of the difference between T_3 and T_4 multiplied by the weight of the fluid passing through the evaporator, and by the specific heat of this fluid.

If we could construct an ideal machine, in which the liquefaction of the gas was automatic, it would be found that the loss of heat in the condenser, measured in thermal units, was exactly equal to the gain of heat in the evaporator. The sensible heat gained and lost by the fluids surrounding the coils in the condenser and evaporator respectively, would be the exact measure of the latent heat of the refrigerating medium, as abstracted in the condenser and returned in the evaporator. It is, however, necessary to change the physical condition of the gas between the evaporator and condenser, so that it can be liquefied in its passage through the latter vessel. Suppose that the pressures in both evaporator and condenser are the same and constant. In order to ensure condensation and liquefaction in the condenser, its temperature would have to be constantly maintained *below* that of the evaporator, a condition of things which is manifestly impracticable since the evaporator is becoming colder with every repetition of the cycle of operations. This difficulty must therefore be met in another way. If we wish to liquefy any gas, it is necessary to bring its molecules closer together, and this can be accomplished either by *increasing* the pressure or by *decreasing* the temperature of the gas, or both. Now, since it is not in this case practicable to reduce the temperature, the only alternative is to raise the pressure by means of the pump already referred to, which draws the gas from the evaporator and delivers it at an increased pressure into the coils of the condenser. But in order to compress a gas, mechanical work must be performed upon it, and this work re-appears in the form of heat. The temperature of the gas after compression is therefore considerably higher than it was at the lower pressure on leaving the evaporator. This heat, in addition to the heat imparted in the evaporator, has to be abstracted and carried away by the cooling action of the water of the condenser.

As stated at the beginning of this Lecture, if we convert a unit weight of any liquid into a gas, we require the addition of a definite amount of heat, and to reconvert this gas into a liquid we require the abstraction of the same amount of heat, the amount being constant for any one liquid at a constant pressure and temperature. All gases do not require the same expenditure of energy to raise them to the same pressure, because they vary in what may be called their compressibility, and some gases occupy a smaller volume than others after an equal amount of compression. Carbon dioxide, for example, according to Regnault, only requires about 75 per cent. of the work necessary to produce the same amount of compression, as air or hydrogen. We can, by experiment, readily determine the exact pressure at which liquefaction will take place at any temperature; and knowing this, the machine can be designed of suitable strength to withstand the necessary pressure.

Owing to the difference in the power required to increase the pressure of different gases, it follows that the amount of heat imparted during compression must with some gases be greater than with others. This fact is of great importance in the selection of a suitable gas, and particularly so if cooling water be scarce. But whatever gas be employed, the pressure necessary to liquefy it must always be increased to a greater or less extent as the temperature of the cooling water rises.

Having considered the principles upon which an evaporative refrigerating machine depends for its action, we are now in a position to examine into the actual question of the interchangeability of heat and work. We can moreover at once establish a coefficient of efficiency for any refrigerating material.

Let, L = Latent heat of evaporation of the refrigerating medium in B.Th.U.

And, H = Heat imparted during compression in B.Th.U.

It follows, from what has been said, that the coefficient of efficiency will be $L \div H$, and, neglecting external losses, wL will represent the heat abstracted in the evaporator, whilst $H + wL$ will equal the heat added to the cooling water of the condenser, where w is the weight in lbs. of the gas entering the condenser in a given time. It is, of course, impossible that a machine could work under such ideal conditions as we have assumed, since there must always be the effect of the high or low temperature of surrounding bodies to determine whether there will be a loss or gain of heat in one or other of the parts of the machine. For instance, it is almost certain that the evaporator will be colder than the atmosphere; in that case, no matter how carefully it may be insulated, there will be some conduction of heat and the net quantity of the heat abstracted from the liquid to be cooled can only be $(w\,L - x)$, where x, is the amount of heat derived from outside sources. The amount of heat imparted to the cooling water in the condenser will therefore be $H + w\,L \pm y$; where y, is the heat lost or gained in the condenser due to the difference in temperature between it and its surroundings.

There is still another correction to be made to the above formula. When, as is usual, the evaporator is maintained at a very low temperature, a certain amount of heat must be imparted to it by the refrigerating liquid itself, as it is entering the evaporator in a comparatively warm condition. Thus, supposing there be t degrees difference in temperature between the condenser and evaporator, a unit weight of the refrigerating liquid will as it were import into the evaporator $t\,s$ thermal units; where s is the specific heat of the liquid in question. Therefore, if W be the weight of refrigerating liquid passing into the evaporator in a given time, the heat abstracted in the evaporator will be represented by the expression :—

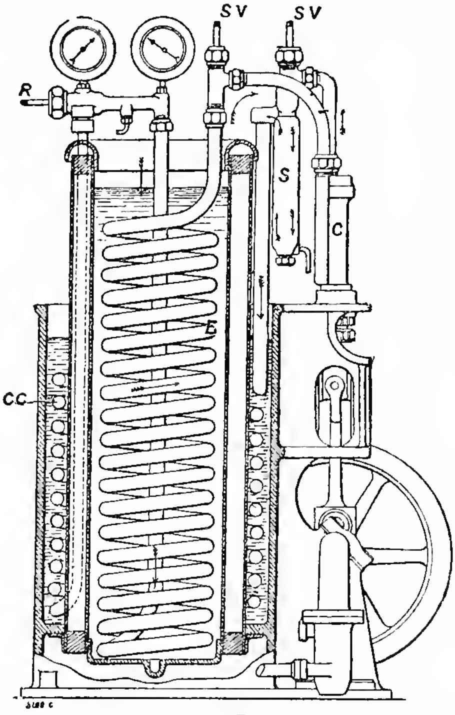
$$w\,L - x - W\,t\,s.$$

Of course, $W\,t\,s$ will not in practice amount to a great deal; but, as Professor Linde has pointed out, it must not be neglected in an exact calculation of the work performed by any refrigerating machine. If there be no leakage, then on the average W will be the same as w.

These formulæ cannot be applied with absolute certainty in practice, owing to the impossibility of making all the necessary corrections due to the gain or loss of heat in the various parts of the machine, and owing to the friction of the gas in constricted passages. But, with care, this gain and loss of heat can be very nearly accounted for in an ordinary machine, as manufactured for commercial purposes and working under the conditions of everyday practice.

Carbon Dioxide Refrigerating Plant.—One method of cooling buildings, &c., on a large scale, is to employ a strong brine obtained by dissolving sodium chloride or common salt in water. This brine is first cooled by passing it through the evaporator of a refrigerating machine, and then circulating it in pipes placed within the chambers which it is desired to cool.

In the accompanying figure all the essential parts are shown of a small refrigerating machine as manufactured by Messrs. J. & E. Hall, of Dartford, for cooling small provision stores, dairies, &c., where the pump may be conveniently driven by a belt. In larger machines, the evaporator is contained in a separate vessel and the compressor is driven by a compound- or triple-expansion engine; but, for the sake of compactness in this case, the evaporator is placed within its condenser, and the intervening space between them is carefully insulated by means of some non-conductor, such as hair felt or slagwool. The coils of



HALL'S CARBON DIOXIDE REFRIGERATING PLANT.

INDEX TO PARTS.

C for Compressor.
S ,, Separator.
CC ,, Condensing Coils.
R ,, Regulator.
G for Gauges.
E ,, Evaporator.
SV ,, Stop Valves.

piping which form the condenser and evaporator are welded into continuous lengths and so connected that all joints shall occur in accessible positions. These and all the other gas joints are made by inserting copper rings turned from the solid metal between a pair of flanges or union coupling. This form of joint has been found very satisfactory. The condenser casing, Corliss frame and bearings for the compression pump, &c., are all made of cast iron in one casting. The compressor C is made of special hard bronze in order to ensure freedom from spongy places, while the suction and delivery valves are identical in shape and size so that they may be interchangeable. The compressors for larger machines are bored out of solid steel forgings. This ensures strength together with sound material. A true bore is also provided for the smooth working of the cupleathers with which the pistons are packed. The gland is made gas-tight by means of two U-leathers fitted over the compressor-rod and glycerine is forced between them under a somewhat greater pressure than that in the compressor. Any leakage which takes place is therefore of glycerine—outwards (which can be collected and used over again) and inwards—which both lubricates the interior of the compressor and fills up the clearance spaces, thereby increasing the efficiency of the machine. The superior pressure of glycerine in the gland is obtained by utilising the pressure in the condenser acting through a small intensifier, similar to those in use in hydraulic installations. Any glycerine which passes into the compresser, beyond what is necessary to fill up the clearance spaces is discharged with the gas through the delivery valves. In order to prevent this glycerine passing into the condenser coils, all the gas is delivered into a separator S and caused to impinge against the sides of this vessel. The glycerine adheres to its sides and drains to the bottom from which it may be drawn off from time to time, thus permitting the dry compressed gas to pass away by an opening at the top of the separator to the evaporator E.

One feature of these machines is the safety valve, which is fitted to the gas circuit immediately above the compressor, so that no harm can be done to the machine even if carelessly started with the stop valves closed. It consists of an ordinary spring safety valve, beneath which is a thin copper disc, designed to burst at a certain pressure. This disc can be made perfectly gas-tight, which could not be so easily accomplished by the spring safety valve alone. The latter only comes into play in the event of a rupture of the copper disc.

Anhydrous Ammonia as a Refrigerating Agent.—The most important advantages possessed by anhydrous ammonia as an agent for cooling purposes are, its freedom from the danger of explosion, its great latent heat and low pressure of vaporisation.* The latent heat of vaporisation of 1 lb. of carbonic acid at 0° F. is 123 units, and of ammonia 555 units, while the respective pressures in lbs. per square inch, at the same temperature, are 310 lbs. in the case of carbonic acid and only 30 lbs. with ammonia. It follows, therefore, that a carbonic acid plant must be constructed to deal with pressures of about 1,000 lbs. per square inch as against only 150 lbs. or so, in an ammonia machine. The exact pressures in each case are directly proportional to the temperature of the condensing water.

As already stated, this agent is more commonly used than any other, and in the United States of America, where refrigeration is applied to an extent unknown elsewhere, the machine generally employed is on the ammonia compression principle. It is similar to the carbonic acid machine in so far as the complete system consists of (1) a compression, (2) a condensing, and (3) an expansion part; moreover, the cycle of operations is exactly the same.

De La Vergne's Refrigerating Plant.—There are many different kinds of ammonia machines in use, but a general description of one of the best known and most extensively applied—viz., the "De La Vergne" as manufactured by Messrs. L. Sterne & Co., Ltd., of the Crown Iron Works, Glasgow, may be taken as a typical example.

In the following figure, A represents the ammonia compressor driven by a steam engine R. The gas which is returned from the expansion coil N, placed in the cooling chamber, enters the cylinder A by the pipe B, and after being compressed therein it is discharged, through the pipe C into a pressure tank D, together with a certain amount of sealing oil. Here, the oil, being heavier than the ammonia gas, naturally falls to the bottom, and the hot ammonia passes from the top of this tank by a pipe E to the condensers F; where, the cooling action of cold water trickling over the pipes causes the gas to liquefy. It then passes through pipes G, G to a header H, and from thence, to a storage tank I, which is simply a receptacle for holding a reserve supply of liquid ammonia. From this tank

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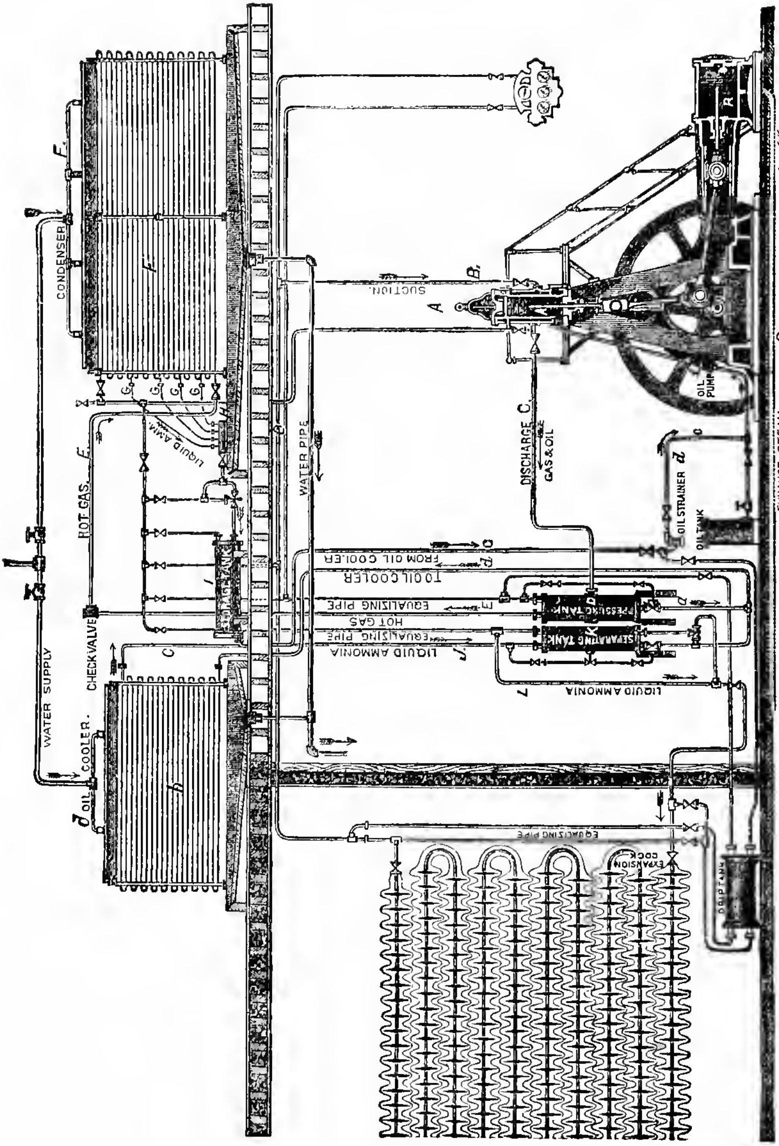
* A liquid with a high latent heat of evaporation need not necessarily be a good refrigerating agent, and *vice-versâ*. What is required is, that its specific heat should be low in proportion to the latent heat of evaporation. Or, we require as great a *difference* as possible between the latent heat of evaporation and the specific heat of the liquid multiplied by the range of temperature in the condenser and refrigerator.

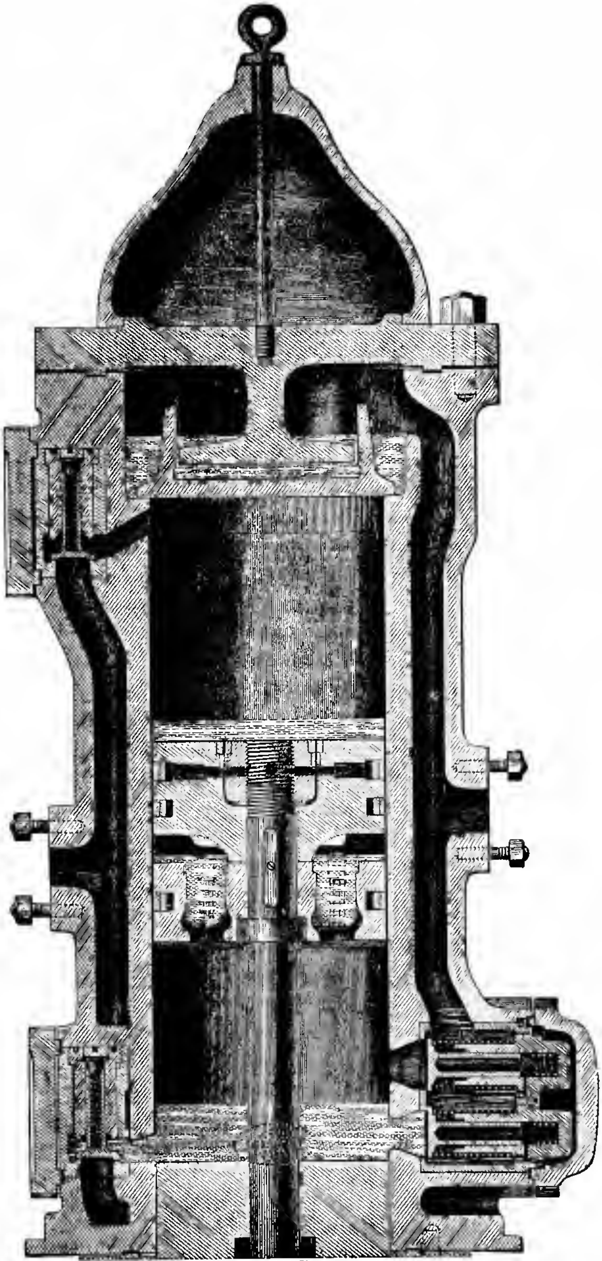


De La Vergne Refrigerating Plant. By L. Sterne & Co., Crown Iron Works, Glasgow.

it is conveyed by a pipe J to a separating vessel K, where any particles of oil that may have been carried over with the liquid are finally separated and the pure liquid ammonia is free to leave it by a pipe L to the expansion coils in the chambers to be cooled. The cock for admitting the ammonia to these coils can be regulated to any degree of minuteness. It thus serves to separate the high pressure from the low pressure part of the apparatus. Hence, the liquid ammonia on passing the expansion cock enters the cooling coils, which are maintained at a low pressure by the pumping action of the compressor. Here, it immediately flashes into gas and by abstracting from its surroundings the heat necessary to cause this change, the temperature of the room is lowered to any desired extent. After having thus done its work in the cooling chamber, the gas is returned to the compressor by a pipe B, to again undergo the same cycle of operations.

The sealing oil passes from the bottoms of the pressure and separating tanks D and K, by the pipes *a* and *d* to the oil cooler *b*; thence, by pipe *c* to the oil strainer *d* and the pipe *e* to the oil pump *f*; by which, it is again circulated through the compressor A.

De La Vergne's Double-Acting Compressor.—The accompanying figure is a section through a "De La Vergne" double-acting compressor, and shows the use of the oil seal. In all ammonia compressors, a certain amount of oil is required for lubricating purposes, and if the compressor be arranged in the ordinary way, the discharge valves at the lower end are placed either on the bottom or at the side, with the result that the oil is discharged *before* the gas. The oil ought, however, to be discharged *after* all the gas is gone; otherwise, re-expansion takes place which would entail a loss of efficiency. In the "De La Vergne" compressor this difficulty has been avoided in the following manner:—At the lower right-hand end of the compressor, two discharge valves are fitted into a side pocket, with the one fair above the other. On the down stroke, either of the valves or both may open until the piston covers the upper one, when only the lower valve is open to the condenser. In the further course of the piston and as soon as the lower valve is also closed, the upper one comes into direct communication with an annular chamber in the piston. This chamber has valves in its bottom side which open into it, as soon as all other inlets on the lower side of the piston are closed. The gas, therefore, first leaves the compressor and then the oil follows, thus permitting no gas to remain in the lower side after the completion of the down stroke. The effect of the oil seal is to make the compressor



De La Vergne's Double-Acting Compressor for his Refrigerator.

work with practically no clearance and thus a maximum of efficiency is obtained. The oil also serves to carry away a considerable amount of the heat of compression and to seal all the valves and stuffing-boxes.

Attention may now be drawn to a few of the details of the above plant. In the first place, it will be noticed that the ammonia condenser is not of the ordinary type where the coils containing the gas are usually submerged in a water tank, but they are of the open or atmospheric type. Here, water is kept constantly trickling over the condenser pipes, and the cooling action is therefore considerably assisted by the evaporation thereof from the surface of the pipes, which enables a maximum of condensation to be effected with a minimum of water supply. It also leaves all the pipes of the condenser open for examination and cleansing. This style of condenser is now coming into extensive use for the condensation of steam in large factories. In the second place, it will be seen that the refrigerating or cooling effect is caused by the direct expansion of the ammonia in pipes placed in the chamber to be cooled. This does away with the unavoidable loss of efficiency due to the use of a supplementary medium such as brine. It, however, necessitates very careful coupling up and jointing all the expansion coils, in order to prevent any leakage of the ammonia gas; more especially, in the case of a large plant where there may be as much as ten or more miles of piping in these cooling coils. In practice, however, these details have been so carefully worked out, that many hundreds of miles of such piping are constantly at work without giving the slightest trouble. Consequently, the old-fashioned method of brine circulation is not now so generally employed except on board ship, where there is a possibility of undue rocking or straining of the pipes and where it is considered advisable to use something that would cause no disagreeable odour in case of a broken pipe or joint.

In applying the refrigerating machine to the manufacture of ice, the simplest method is to place the expansion or cooling coils in a tank filled with brine or other non-congealable liquid, while the water to be frozen is placed in moulds of suitable size, which are then inserted into this brine until frozen. The purpose served by the brine in this case is to convey the heat from the water to the cooling pipes. It is therefore generally kept in slow circulation in order to ensure that the temperature shall be as uniform as possible throughout the tank. If ordinary well water be placed in the moulds, the resulting ice will contain so much air that it will be turned out of a milky white and opaque colour; but if the water whilst in the process of freezing be kept in slow motion by means of agitators, this air escapes, and a clear glassy ice is the result.

Another method of obtaining clear ice is to use distilled water. This has also the advantage of getting rid of any objectionable matter which might be in solution.

The Linde System of Refrigeration.—This system was first introduced into Germany in the year 1875 by Professor Linde, who was then one of the teaching staff at Munich University. In this country, however, prior to 1888, the principal cold-producing machinery, as manufactured for both land and marine purposes, was the simple cold air machine, in which refrigeration is produced by the compression, cooling when under compression by means of water and subsequent expansion of ordinary atmospheric air. These machines, although simple in construction and giving very good results, possess the disadvantage of requiring a large amount of power to work them in comparison with those employing more efficient refrigerating agents. Consequently, the former method has now very largely given way to one or other of the latter, of which the Linde system is one of the most successful, seeing that over 3,000 of these machines have been constructed up to September, 1897, representing an output, as rated by the capability of producing 69,200 tons of ice every twenty-four hours. In America the largest machine turned out upon any system is rated at 500 tons of ice per twenty-four hours.

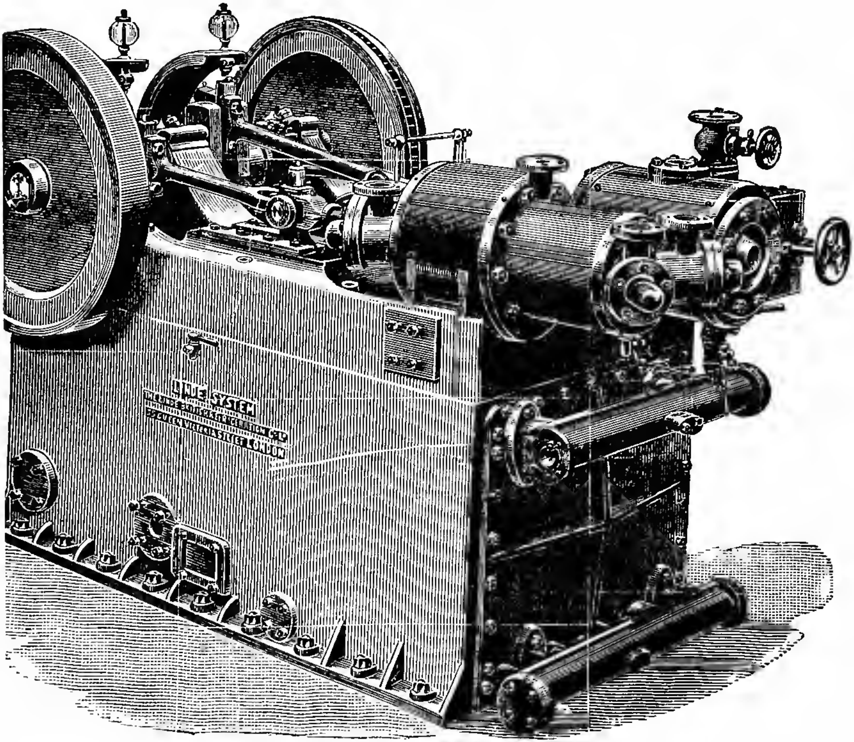
The Linde System of Refrigeration is identical in principle, and only differs in mechanical details from the De La Vergne previously described. It is, therefore, based on the evaporation of liquid anhydrous ammonia and the subsequent liquefaction thereof by means of mechanical compression, together with the cooling of the vapour thus formed, so as to enable it to be used over and over again. As will be seen from the accompanying illustration, the self-contained motive-power plant, as chiefly used on board ship, consists of a horizontal steam engine on the right, with a horizontal duplex compressor pump to the left, and an ammonia condenser in the sole plate. As far as the compressor is concerned, the chief differences between the Linde and the De La Vergne systems are:—

(1) That in the former a horizontal compressor is used instead of a vertical one in the latter case.

(2) That a special oil (not susceptible to change at any temperature attainable by the machine, which does not contain any acid or other deleterious matter, and which does not saponify when brought into contact with ammonia is used solely for lubricating purposes. Whereas, the oil used in the De La

Vergne system serves not only as a lubricant to the working parts, but also to partly carry away the heat of compression, and, further, to fill up the clearance spaces, as well as to seal the valves, glands, &c., so as to prevent the escape and consequent inconvenient smell of ammonia.

(3) In the Linde system a small quantity of liquid ammonia is introduced into the compressor during each suction stroke for the purpose of cooling the vapour of ammonia. This liquid ammonia evaporates during compression, and thus the heat due to compression, which would otherwise appear as sensible heat, is thereby absorbed and rendered latent in producing the change in the physical state of the liquid. The curve of compression is thus kept down as nearly as may be to the isothermal line, and


LINDE SYSTEM
THE LINDE BRITISH REFRIGERATION Co Ld
35 QUEEN VICTORIA STREET LONDON

Ship Refrigerating Machinery on the Linde System.
By The Linde British Refrigerating Company.
]

the power required for compression is to this extent correspondingly reduced.

Apparatus for Transmitting the Cold Produced to the Chambers requiring Refrigeration.—The following general principles are adopted for transferring the cold generated by the refrigerating machinery to the chambers or rooms requiring to be cooled.

First Method.—An uncongealable solution of salt (chloride of sodium) in water is reduced by the refrigerating machine to a low temperature, and this liquor acts as transmitter of cold in one of the following methods:—

(*a*) The cold brine is constantly circulated from the brine refrigerator through pipes placed in the refrigerated chambers, and returned to the brine cooler. The result is that not only is heat abstracted from the air of the refrigerated rooms, but also a large degree of the moisture which may be present in them. This moisture is condensed on the exterior of the brine pipe systems either in the form of condensed water or hoar-frost. Suitable drip trays are provided, in order to prevent this moisture from falling upon the contents of the rooms. The circulation of air with this system is a moderate one, being produced merely by the differences between the temperatures prevailing near the brine-pipes and those in the lower parts of the rooms.

(*b*) The brine is cooled in a shallow rectangular open tank containing the evaporator coils. On the tank is mounted a number of slowly revolving transverse shafts, and on each shaft is fixed a number of parallel discs, partly immersed in the brine, the entire apparatus being placed in an insulated passage through which an air current is continually passed by a fan, in a direction parallel to the revolving discs. It will be seen, that as the discs revolve and are kept covered by a film of the refrigerated brine, the air passing between the disc-spaces becomes cooled, and produces a low temperature in any chamber or room into which it may be conducted through properly arranged air-trunks. As a rule the air is always taken back from the cold rooms, passed over the discs and returned to the cold rooms, and any required amount of fresh air is introduced by means of adjustable openings in the air-trunks, communicating with the outer atmosphere. In this instance, also, moisture may be removed from the refrigerated rooms and deposited in the brine contained in the trough. No accumulation of frost can take place, and the refrigerated surfaces are always perfectly active. The circumstance of all moisture being deposited in the brine necessitates either a periodical loss of the same or its re-concentration. The fan produces a very effective air circulation within the rooms to be cooled. This in most cases is extremely desirable, and, as will be readily understood, produces the most beneficial results.

Second Method.—Instead of using an uncongealable liquid as a bearer of cold, the refrigerator coils (in which the vaporisation of the liquid anhydrous ammonia takes place) are sometimes constructed with extra large surfaces, and placed either in the upper part of the rooms to be cooled, or in a separate chamber. In the latter case, a fan constantly circulates the air between this chamber and the refrigerated rooms. This is the system generally adopted on board ships, and has been found to be in all respects most satisfactory. In cases where the air temperature is not sufficiently high to cause a complete removal of the snow deposited on the ammonia-coils, the snow is thawed by the ammonia vapours themselves, the evaporator-coils being for the time used as a condenser. Occasionally the snow is thawed by a current of hot air taken from the outside.

Although all of these methods have been applied on an extensive scale, the system most strongly recommended in cases where its application is possible is the combination of revolving discs immersed in brine. There are no brine or ammonia pipes in the rooms; whilst the rapid air circulation by the fan is easily managed, and has been found in most cases to be requisite for obtaining a satisfactory result as to purity, dryness, and equable temperature in all the rooms.

Where circumstances require the refrigerated rooms to be at a distance from the refrigerating machine, it is generally most convenient to place bundles of brine pipes in each room; but even in such a case, in the event of a small amount of motive power being available close to such rooms, the system of revolving discs and fans can be readily applied, the brine being cooled in a refrigerator near the compressor, and conveyed to and from the disc tanks through insulated pipes.

A large beef-chilling plant on the Linde system was erected in the beginning of 1890 at the Woodside lairage of the Mersey Dock and Harbour Board. It is capable of chilling 660 carcases of beef, each weighing about 9 cwts., from 90 degrees Fahr. to 33 degrees in 17 hours. It consists of a horizontal compound tandem jet-condensing steam engine, which drives a double-acting Linde compressor at the rate of about 65 revolutions per minute, when supplied with steam at 120 lbs. pressure from a marine type boiler. The air-cooling apparatus consists of two disc tanks, placed above the chill rooms, at one end. Each disc system has its own fan, which draws the air from the top of each of the chill rooms, passes it over the discs, and drives it into the rooms at the opposite end to that from which it is withdrawn.

The ammonia condenser is placed in the compressor-room, and is supplied with cooling water by a pump which takes its supply from a well, fed with the drainage water from the Mersey Tunnel. After passing through the ammonia condenser the water is used in the condenser of the steam engine. There are six chill rooms, each about 55 ft. long by 14 ft. wide, and about 13 ft. high. The walls of the rooms are built of brick, with air spaces. The floors are cement, and the ceilings are timber, covered with a layer of fine ashes. The air-cooling apparatus is contained in an insulated casing, which is so arranged as to cause the air to come in contact with the cooled surfaces of the discs.

The following is a list of some of the books and papers on Hydraulics and Hydraulic Machinery :—

Hydraulic Machinery, by R. Gordon Blaine. (E. & F. N. Spon, Ltd., London, 1897.)

Recent Hydraulic Experiments. Paper by Major A. Cunningham. Proc. Inst. C.E., vol. lxxi., p. 1.

Hydraulic Appliances at the Forth Bridge Works. Paper by E. W. Moir. Proc. Inst. C.E., vol. xci., p. 402.

The 160-Ton Hydraulic Crane at Malta Dockyard Extension Works. Paper by C. & C. H. Colson. Proc. Inst. C.E., vol. cxiv., p. 289.

On Machine Tools and other Labour-Saving Appliances Worked by Hydraulic Pressure. Paper by R. H. Tweddell. Proc. Inst. C.E., vol. xxiii., p. 64.

The Barry Dock Works, including the Hydraulic Machinery and the Mode of Tipping Coal. Paper by J. Robinson. Proc. Inst., C.E., vol. ci., p. 129.

The Distribution of Hydraulic Power in London. Paper by E. B. Ellington. Proc. Inst. C.E., vol. xciv., p. 1.

Hydraulic-Power Supply in London. Paper by E. B. Ellington. Proc. Inst. C.E., vol. cxv., p. 220.

Forging by Hydraulic Pressure. Paper by R. H. Tweddell. Proc. Inst. C.E., vol. cxvii., p. 1.

Hydraulic-Power Supply in Towns. (Glasgow, Manchester, Buenos Ayres, &c.) By Edward B. Ellington, M. Inst. M.E. (Proc. Inst. Mechanical Engineers, Glasgow Meeting, 1895).

Water-Pressure Engines for Mining Purposes. Paper by H. Davey, M. Inst. C.E. Proc. Inst. M.E., 1880, p. 245.

Hydraulic Flanging of Steel Plates. Paper by Easton & Anderson. Proc. Inst. M.E., 1882, p. 518.

Portable Hydraulic Drilling Machine. Paper by M. Berrier-Fontaine. Proc. Inst. M.E., 1887, p. 72.

Lifts in the Eiffel Tower. Paper by A. Ansaloni. Proc. Inst. M.E., 1889, p. 350.

Hydraulic Packing Presses. Paper by C. Hopkinson. Proc. Inst. C.E., vol. xcix., p. 275.

The Treatment of Steel by Hydraulic Pressure and the Plant Employed for the purpose. Paper by W. H. Greenwood. Proc. Inst. C.E., vol. xcviii., p. 83.

Hydraulic Propulsion. Paper by S. W. Barnaby. Proc. Inst. C.E., vol. xxvii., p. 83.

A Hydraulic Pumping Engine. Paper by H. D. Pearsall. Proc. Inst. C.E., vol. cvi., p. 292.

Hydraulic Work in the Irawadi Delta. Paper by R. Gordon. Proc. Inst. C.E., vol. cxiii., p. 276.

Centrifugal Pumps, Turbines, and Water Motors, by C. H. Innes, M.A. (The Technical Publishing Co., Ltd., Manchester.)

Lifting and Hauling Appliances in Portsmouth Dockyard. Paper by J. T. Corner. Proc. Inst. M.E., 1892, p. 295.

Hydraulic Power and Hydraulic Machinery, by Henry Robinson. (Chas. Griffin & Co., London, 1896.)

Hydraulic Motors, Turbines, and Pressure Engines, by G. R. Bodmer. (Whittaker & Co., London, 1889.)

A Treatise on Hydraulics, by M. Merriman. (John Wiley & Sons, New York ; Chapman & Hall, London, 8th Edition, 1903.)

Hydrostatics and Elementary Hydrokinetics, by G. M. Minchin. (Oxford Press, Oxford, 1892.)

Hydraulic Machinery for Glasgow Harbour Tunnel, as Constructed and Erected by the Otis Elevator Co., New Yonkers, New York. See *Engineering,* May and June, 1895.

Paper on *Refrigerating Apparatus,* by Prof. Carl Linde, of Munich. *Journal of the Society of Arts,* 9th March, 1894.

Paper on *Experiments on a Two-Stage Air Compressor,* by John Goodman, Wh.Sc. Proc. Inst. C.E., vol. cxxviii., and also *The Practical Engineer,* 30th July, 1897.

Compend of Mechanical Refrigeration, by J. E. Siebel. (H. S. Rich & Co., Chicago, 1895.) In this book there are several important tables and other valuable information regarding refrigerators.

Adaptation of Hydraulic Power in the Manufacture of Iron and Steel. Paper by James L. Biggart. *Journal of the West of Scotland Iron and Steel Institute,* 1893.

Application of a System of Combined Steam and Hydraulic Machinery to the Loading, Discharging, and Steering of Steamships. Paper by A. Betts Brown, F.R.S.E. Proc. Inst. Nav. Arch., 1890.

The Theory and Practice of Mechanical Refrigeration. Paper by T. R. Murray, Wh. Sch. Proc. Inst. Eng. and Shipbuilders Scot., 1897-98.

The Mechanical Production of Cold. Howard Lecture by Prof. J. A. Ewing, F.R.S. Delivered Feby. 4, 1897, before The Society of Arts, London. See their *Journal* or *The Practical Engineer* for Dec., 1897, and Jan., 1898.

Pumping Machinery, by Henry Davey, M.Inst.C.E. (Charles Griffin & Co., London, 1903.)

Pneumatic Tools.—The pneumatic tools of the present day, form most useful appliances in up-to-date engineering shops. They are adaptable to many kinds of work of which the following are a few examples:—

Classes of Pneumatic Hammers.—These may be divided into two classes—(*a*) the *valveless hammer*, and (*b*) the *valve hammer*.

(*a*) The *valveless hammer*, as its name implies, has no separate valve, but its striking piston or plunger does duty as a valve, and regulates the proper admission of air to alternate ends of the working cylinder.

(*b*) The *valve hammer* has a reciprocating valve placed either horizontally or vertically, which controls the admission of the air to the working cylinder and also the exhaust therefrom. The valve hammer is the type in general use throughout engineering shops and shipyards, since the valveless hammer is too rapid and light in its blow. The estimated number of strokes per minute of the "valve hammer" is from 700 to 2,000; whereas, in the case of the "valveless hammer," they may vary from 10,000 to 18,000 strokes per minute.

Uses and Capabilities of Pneumatic Hammers.—When used for caulking, they will perform the work of three to four skilled hand caulkers. When used in the foundry for scarfing or dressing castings, they will do the work of from four to five men. In general chipping, they will do the work of at least three men. They are also used in stone dressing, lettering, and carving, with considerable saving in time and cost.

The Cleveland Hammer.*—A further development of pneumatic chipping and caulking hammers, is found in the hand pneumatic riveting hammer. We have here chosen to illustrate and describe the "Cleveland Hammer," which is very strong and at the same time simple in construction, for it has only one solid steel valve.

Action of the Cleveland Hammer.—Referring to the diagram, with index to parts, of this hammer, it will be seen, that it consists of the handle, H, which is connected to the barrel, B, by the coupling sleeve, C S. Inside this barrel is placed the plunger, P, whilst between the end of the barrel and the handle is secured a valve block button, $V B_1$, and valve block, $V B_2$, in which the valve, V, works. Compressed air is admitted to the interior of the hammer by the action of the trigger or latch, L, upon the latch valve pin, V P, which in turn acts upon the latch valve, L V, and valve spring, V S. The compressed air when admitted, passes through a hollow chamber in the handle, H, and acts upon the valve, V, thus rapidly moving the plunger, P, backwards and forwards. A rivet set, R S, is placed in the end of the hammer or tool nose, T N, and held by a spring holder-on, S H. The rivet set, R S, must be placed hard on the rivet, before the compressed air is admitted, otherwise the plunger and rivet set would be blown out of the hammer.

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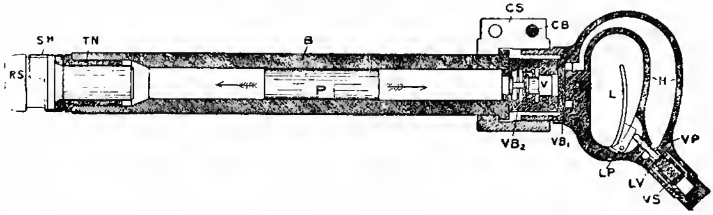
* We are indebted to Messrs. John Turnbull, Jun., & Sons, Hydraulic and Pneumatic Engineers, Glasgow, for the original figures from which the illustrations in this article were produced.



SECTIONAL VIEW OF A "CLEVELAND" PNEUMATIC HAMMER.

INDEX TO PARTS.

H	for Handle.	**C S**	for Coupling sleeve.
V P	,, Latch valve pin.	**B**	,, Barrel.
V C	,, Latch valve casing.	**P**	,, Plunger.
V S	,, Latch valve spring.	**T N**	,, Tool nose.
L V	,, Latch valve.	**V B$_2$**	,, Valve block.
L P	,, Latch pin.	**S H**	,, Spring holder-on.
L	,, Latch.	**R S**	,, Rivet set.
V B$_1$	,, Valve block button.	**C B**	,, Cap bolt.
V	,, Valve.		

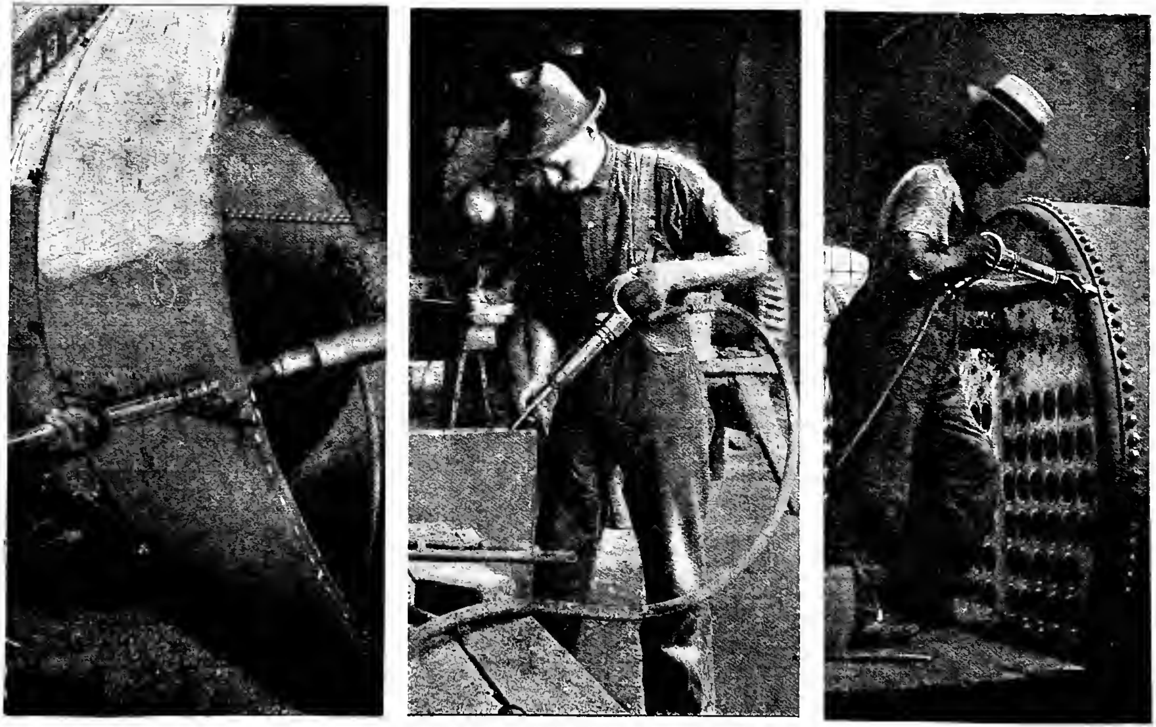
Capabilities of the Cleveland Hammer.—With one of these riveting hammers a man can fix and finish 100 of ½-inch rivets, 90 of ⅝-inch, 80 of ¾-inch, and 70 of ⅞-inch per hour.

Sizes and Speeds of Cleveland Hammers.—These hammers are made in thirteen sizes to take in all classes of riveting work. The strokes of the various sizes are as follows:—

5-inch stroke, ...	800 strokes per minute.
6 ,, ,, ...	775 ,, ,,
7 ,, ,, ...	750 ,, ,,
8 ,, ,, ...	725 ,, ,,
9 ,, ,, ...	700 ,, ,,

This allows for a variety of work ranging from ¼-inch cold rivets up to and including 1½-inch hot rivets.

Pneumatic Holder-up.—When riveting with pneumatic hammers, it is always best to use a pneumatic holder-up, so that the head of the rivet shall be kept close up to the plate. Owing to the rapid succession of blows given by these hammers, the



Riveting, Chipping, and Caulking with the Cleveland Pneumatic Hammer.

The pneumatic holder-up consists of a piston placed inside a cylinder, into which compressed air is admitted. The end of the piston being kept up to the rivet head by the air pressure in its cylinder.

Pneumatic Plant.—The following plant would be required for working a number of these hammers :—

1. Air compressor, either belt, steam, or electrically driven.

2. An air receiver, to hold not less than the output of the air compressor for one minute. This receiver forms an important part of the air compressing plant. It receives the air in pulsations from the air compressor, until a pressure of about 100 lbs. per square inch is obtained, and then delivers it to the tools in a steady, even flow. It also receives any moisture which may be in the compressed air, and provision must be made in the receiver for drawing off this moisture.

3. Lines of wrought-iron piping, for conveying the compressed air to convenient parts of the workshop. These pipes should be fitted with air strainers, in order to clear the compressed air, as far as possible, from grit, &c.

4. Lengths of flexible tubing or rubber-lined hose pipes, to form the connection between the line of wrought-iron air pipes and the pneumatic tools.

5. Pneumatic tools to perform the work desired.

Advantages of Pneumatic Plant for Riveting, Chipping, and Caulking.—1. This system is perhaps the best of all when applied to suitable machines, since compressed air, on account of its comparatively low pressure and weight, is more easily distributed than water.

2. There is not much risk from freezing, which is an advantage where plant, pipes, and tools are of necessity exposed to cold weather. In cases, however, where the compressed air had been allowed to contain a certain percentage of moisture in it, there has been a little trouble in the winter time, due to ice forming in the pipes.

3. The cost of a complete pneumatic plant, is said to be not quite so much as that for a corresponding hydraulic one.

4. As previously mentioned in this article, much more and quite as good work can be done, than by ordinary hand riveting, chipping, and caulking.

LECTURE VI.—QUESTIONS.

1. Explain the fundamental principles upon which a refrigerating machine works. Note specially what becomes of the different quantities of heat generated and absorbed.

2. Sketch the essential parts of a refrigerator, and describe its action.

3. What are the advantages which a vapour possesses over a permanent gas, such as air, for refrigerative purposes?

4. What are the requirements of an economical medium for use in a refrigerator?

5. Sketch and explain the plant required for producing cold by means of carbon dioxide.

6. Explain the reasons that have led to the adoption of anhydrous ammonia in most modern refrigerators, and mention some of the properties of this vapour.

7. Sketch and describe any well-known arrangement for refrigerating, using anhydrous ammonia.

8. Explain and illustrate some of the ways of communicating cold to a chamber from a refrigerator.

9. Sketch a longitudinal section of a pneumatic hammer and describe concisely its construction, action, and uses. State the advantages which are claimed for this hammer in riveting, chipping, and caulking work. Compare the work done by it with ordinary hand riveting, chipping, and caulking.

10. Describe and sketch any refrigerating system you may be acquainted with. Give the temperatures at the different stages. Give particulars of any parts which are subjected to pressure, and state the pressures to which these parts are subjected. (*From the Board of Trade Examination for Extra Chief Marine Engineers, Oct.*, 1904.)

11. Describe, with sketches, the hydraulic apparatus in connection with a hydraulic riveter, and investigate what effect a long line of pipe of small diameter between the accumulator and the riveter has on the pressure applied to close the rivet. Sketch, roughly, an indicator diagram for the riveter ram, and trace on it the various losses, &c.
(C. & G., 1904, H., Sec. C.)

N.B.—*See Appendices B and C for other questions and answers.*

APPENDICES.

APPENDIX A (p. 224 to p. 229).

(i.) General Instructions by The Board of Education for their Examinations on Applied Mechanics.

(ii.) General Instructions by The City and Guilds of London Institute for their Examination on Mechanical Engineering.

(iii.) Rules and Syllabus of Examinations by The Institution of Civil Engineers for Election of Associate Members.

APPENDIX B (p. 230 to p. 258).

(i.) Board of Education's Exam. Papers in Applied Mechanics, Stages 2 and 3. The City and Guilds of London Institute's Honours Exam. Papers in Mechanical Engineering. And, The Institution of Civil Engineers' Exam. Papers in Hydraulics, arranged in the order of the Lectures.

(ii.) All new Answers to Questions for the respective Lectures are tabulated under the two main headings—*either* Board of Education and City and Guilds *or* Institution of Civil Engineers.

APPENDIX C (p. 259 to p. 262).

The latest Exam. Papers pertaining to Hydraulics and set by the governing bodies enumerated under Appendix A.

APPENDIX D (p. 263 to p. 268).

Tables of Constants, Logarithms, Antilogarithms, and Functions of Angles.

APPENDIX A.

MAY EXAMINATION ON SUBJECT VIIA. APPLIED MECHANICS.*

BY THE BOARD OF EDUCATION SECONDARY BRANCH,
SOUTH KENSINGTON, LONDON.

GENERAL INSTRUCTIONS.

If the rules are not attended to, your paper will be cancelled.

Immediately before the Examination commences, the following REGULATIONS *are* TO BE READ TO THE CANDIDATES.

Before commencing your work, you are required to fill up the numbered slip which is attached to the blank examination paper.

You may not have with you any books, notes, or scribbling paper.

You are not allowed to write or make any marks upon your paper of questions, or to take it away before the close of the examination.

You must not, under any circumstances whatever, speak to or communicate with one another, and no explanation of the subject of examination may be asked or given.

You must remain seated until your papers have been collected, and then quietly leave the examination room. None of you will be permitted to leave before the expiration of one hour from the commencement of the examination, and no one can be re-admitted after having once left the room.

Your papers, unless previously given up, will all be collected at 10 o'clock.

If any of you break any of these rules, or use any unfair means, you will be expelled, and your paper cancelled

Before commencing your work, you must carefully read the following instructions :—

Candidates who have applied for examination in the Elementary Stage must confine themselves to that stage. Candidates who have not applied to take the Elementary Stage may take Stage 2, or Stage 3, or, if eligible, Honours, but they must confine themselves to one of them.

Put the number of the question before your answer.

You are to confine your answers *strictly* to the questions proposed.

Such details of your calculations should be given as will show the methods employed in obtaining arithmetical results.

A table of logarithms and functions of angles and useful constants and formulæ is supplied to each candidate. (See end of Appendix to this Book.)

The examination in this subject lasts for three hours.

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* *See Appendix C for the latest Exam. Papers.*

City and Guilds of London Institute.

DEPARTMENT OF TECHNOLOGY.

TECHNOLOGICAL EXAMINATIONS.

46.—MECHANICAL ENGINEERING.*

Honours Grade (Written Examination).

Instructions.

The Candidate for Honours must have previously passed in the Ordinary Grade, and is required to pass a Written and Practical Examination. He is requested to state, on the Yellow Form, whether he has elected to be examined in A, Machine Designing, or in B, Workshop Practice (*a*) *Fitting*, (*b*) *Turning*, (*c*) *Pattern-making*. Candidates in Machine Designing must forward their work to London not later than May 6th,† and in Workshop Practice not later than May 13th.†

The number of the question must be placed before the answer in the worked paper.

The Candidate is at liberty to use divided scales, compasses, set squares, calculators, slide rules, and mathematical tables.

Five marks extra will be awarded for every answer worked out with the slide rule, provided the method of working is explained.

The maximum number of marks obtainable is affixed to each question.

Three hours allowed for this paper.

The Candidate is not expected to answer more than *eight* of the following questions, which must be selected from *two* sections *only*.

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* The questions in Sections A and C may be answered from Vols. I. to V. of my *Text-Book on Applied Mechanics*. Section B is printed in my *Text-Book on Steam and Steam Engines*. The Questions for the Ordinary Grade are printed at the end of my *Elementary Manual on Applied Mechanics*.

* *See Appendix C for the latest Exam. Papers.*

† These dates are only approximate, and subject to a slight alteration each year.

The Institution of Civil Engineers.

EXTRACTS FROM RULES AND SYLLABUS OF EXAMINATIONS FOR ELECTION OF ASSOCIATE MEMBERS.

Note.—Engineers who desire to enter for the A.M.Inst.C.E. examinations should write *at once* to the Secretary, Great George Street, Westminster, S.W., for the complete Rules, Syllabus, and Application Forms. They will find all the questions relating to the above mentioned subjects which have been set since these examinations commenced in 1897 in my **Text-Books.**

The following extracts are simply printed here to show how far my hooks upon *Applied Mechanics and Mechanical Engineering; Steam and Steam Engines, including Turbines and Boilers;* as well as *Magnetism and Electricity* (including Munro & Jamieson's *Pocket-Book of Electrical Rules and Tables*), together with my "*Correspondence System of Electrical and Mechanical Engineering Science, as taught by Exercises, Drawings, and Instructions*" cover the Scientific and Practical Knowledge demanded by the Institution, under Part II., **Section A (1, 2, 3*a*, and 3*b*)**, as well as **Section B** under Group i., Theory of Heat Engines; Group ii., Hydraulics and Theory of Machines; and Group iii., Applications of Electricity.

Note for Students.—In the regulations for Students one of the subjects which may be selected is that of Elementary Mechanics of Solids and Fluids, for which see my *Elementary Applied Mechanics* when studying this subject. Also, see my *Elementary Magnetism and Electricity* book when reading that part of the Elementary Physics for admission of Students, or correspond with me *re* those two subjects.

PART II.*—*Scientific Knowledge.*

SECTION A.

1. **Applied Mechanics** (one Paper, *time allowed*, 3 *hours*).
2. **Strength and Elasticity of Materials** (one Paper, *time allowed*, 3 *hours*).

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* Candidates may offer themselves for examination in Sections A and B of Part II. together; or they may enter for Section A alone, and, if successful, may take Section B at a subsequent examination. In the latter case, however, such candidates will not be allowed to present themselves for examination in Section B unless or until they are actually occupied in work as pupils or assistants to practising Engineers. The Council may permit Candidates who have attempted the whole of Part II. at one examination, and have failed in Section B only, to complete their qualification by passing in that section at a subsequent examination, subject to their being then occupied as above stated.

3. *Either* (*a*) Theory of Structures.
or (*b*) Theory of Electricity and Magnetism (one Paper, *time allowed*, 3 *hours*).

SECTION B.

Two of the following nine subjects—not more than one from any group (one Paper in each subject taken, *time allowed*, 3 *hours for each Paper*):—

Group i.	Group ii.	Group iii.
Geodesy.	Hydraulics.	Geology and Mineralogy.
Theory of Heat Engines.	Theory of Machines.	Stability and Resistance of Ships.
Metallurgy.	Thermo- and Electro-Chemistry.	Applications of Electricity.

Mathematics.—The standard of Mathematics required for the Papers in Part II. of the examination is that of the mathematical portion of the Examination for the Admission of Students, though questions may be set involving the use of higher Mathematics.

The range of the examinations in the several subjects, in each of which a choice of questions will be allowed, is indicated generally hereunder:—

SECTION A.

1. Applied Mechanics:—

Statics.—Forces acting on a rigid body; moments of forces, composition, and resolution of forces; couples, conditions of equilibrium, with application to loaded structures. The foregoing subjects to be treated both graphically and by aid of algebra and geometry.

Hydrostatics.—Pressure at any point in a gravitating liquid; centre of pressure on immersed plane areas; specific gravity.

Kinematics of Plane Motion.—Velocity and acceleration of a point; instantaneous centre of a moving body.

Kinetics of Plane Motion.—Force, mass, momentum, moment of momentum, work, energy, their relation and their measure; equations of motion of a particle; rectilinear motion under the action of gravity; falling bodies and motion on an inclined plane; motion in a circle; centres of mass and moments of inertia; rotation of a rigid body about a fixed axis; conservation of energy.

2. Strength and Elasticity of Materials:—

Coefficients of elasticity; elastic resistance to tension, compression, shearing, and torsion; uniform and varying stress; moment of stress and of resistance in beams of various sections; distribution of shearing stress in beams; stresses suddenly applied and effects of impact; buckling of struts; effect of different end-fastenings on their resistance; combined strains; calculations connected with statically indeterminate problems, as beams supported at three points, &c.; limit of elasticity, yield-point, and ultimate resistance of various materials as tested: the plastic state; principal forms of testing machines and of appliances used in measurements for the determination of coefficients of elasticity; calculation of extension, deflection, buckling, &c., within elastic limits, and of ultimate strength and ductility for ordinary materials of construction.

3. (*a*) Theory of Structures:—

Graphic and analytic methods for the calculation of bending moments and of shearing forces, and of the stresses in individual members of framework structures loaded at the joints; plate and box girders; incomplete and redundant frames; theory of continuous girders and principal methods of calculation; travelling loads; riveted and pin-joint girders; rigid and hinged arches; strains due to weight of structures; theory of earth-pressure and of foundations; stability of masonry and brickwork structures.

3. (*b*) Theory of Electricity and Magnetism:—

Electrical and magnetic laws, units, standards, and measurements; electrical and magnetic measuring instruments; the theory of the generation, storage, transformation, and distribution of electrical energy; continuous and alternating currents; arc and incandescent lamps; secondary cells.

SECTION B.

Group i. Theory of Heat Engines:—

Thermodynamic laws; internal and external work; graphical representation of changes in the condition of a fluid; theory of heat engines working with a perfect gas; air- and gas-engine cycles; reversibility, conditions necessary for maximum possible efficiency in any cycle; properties of steam; the Carnot and Clausius cycles; entropy and entropy-temperature diagrams, and their application in the study of heat engines; actual heat engine cycles and their thermodynamic losses; effects of clearance and throttling; initial condensation; testing of heat engines, and the apparatus employed; performances of typical engines of different classes; efficiency.

Group ii. Hydraulics:—

The laws of the flow of water by orifices, notches, and weirs; laws of fluid friction; steady flow in pipes or channels of uniform section; resistance of valves and bends; general phenomena of flow in rivers: methods of determining the discharge of streams; tidal action; generation and effect of waves; impulse and reaction of jets of water; transmission of energy by fluids; principles of machines acting by the weight, pressure, and kinetic energy of water; theory and structure of turbines and pumps.

Theory of Machines:—

Kinematics of machines; inversion of kinematic chains; virtual centres; belt, rope, chain, toothed and screw gearing; velocity, acceleration and effort diagrams; inertia of reciprocating parts; elementary cases of balancing; governors and flywheels; friction and efficiency; strength and proportions of machine parts in simple cases.

Group iii. Applications of Electricity:—

Theory and design of continuous- and alternating-current generators and motors, synchronous and induction motors and static transformers; design of generating- and sub-stations and the principal plant required in them; the principal systems of distributing electrical energy, including the arrangement of mains and feeders; estimation of losses and of efficiency; principal systems of electric traction; construction and efficiency of the principal types of electric lamps.

☞ Candidates should see, that all their "Forms" are *duly completed* and *passed* by the Council of the Institution of Civil Engineers, Great George Street, Westminster, S.W., *before* 1st January for the February Examination, and *before* the 1st September for the October Examination. Candidates should, therefore, apply to the Secretary for the "Forms," *at least six* months *before* these Examinations, to give them time to make due and proper Application, and to thoroughly *Revise* the subjects upon which they are to be examined with an experienced **Guide** and Tuition by Correspondence.

Examinations Abroad.—The papers of the *October* Examination *only* will be placed before accepted Candidates in India and the Colonies. To enable the Secretary to make arrangements for the Application Forms and Fees, &c., of these Candidates, their Forms, &c., *must* be in the Secretary's hands, *before* the 1st June preceding the October Examinations.

N.B.—*See Appendix C for the latest Exam. Papers in Hydraulics.*

APPENDIX B.

Lecture I.—Ordinary Questions.

1. Sketch any form of water accumulator with which you are acquainted, and explain the advantages it possesses over the arrangement in which the pumps pump water directly into the mains. In an accumulator the total force in lbs., due to cup leather friction, may be taken equal to ·047 *p d*, where *p* is the pressure in lbs. per square inch and *d* is the diameter of the ram in inches. The ascending pressure is 1,250 lbs. per square inch, and the diameter of the ram is 6 inches. Calculate the descending pressure, and deduce the efficiency of the accumulator as a storer of energy. (C. & G., H., Sec. C, 1903.)

2. Define what is meant by the terms coefficient of velocity, contraction and discharge, in connection with the flow through orifices. A vessel containing water and having a circular orifice, 1 inch in diameter, in one of its sides is supported in such a way that when a jet of water issues from the orifice the displacing force can be very accurately measured. With a constant head of 2¼ feet of water above the orifice, the displacing force (at the centre of the orifice) was found to be ·9 lb. and the discharge 15½ gallons per minute. Determine the three coefficients. (C. & G., H., Sec. C, 1903.)

3. When suction pumps are run at too high a speed, "shock" or "separation" takes place. Explain, as clearly as possible, the precise action and upon what the limiting speed depends. A single-acting suction and lift plunger pump, whose piston has a diameter of 12 inches and a stroke of 2 feet, with a suction pipe 33 feet long and 10 inches diameter, has a lift of 20 feet—the barometric head being 34 feet of water. Assuming that the piston moves in a simple harmonic manner, find the limiting number of revolutions per minute of the crank-shaft below which separation will not take place at the commencement of the stroke. (C. & G., H., Sec. C, 1903.)

4. Describe, with the aid of sketches, any hydraulic pumping engine with which you are acquainted in which the delivery is the same for both strokes. Show, in particular, the arrangement of valves. (C. & G., H., Sec. C, 1903.)

5. Calculate in tons per minute, the quantity of water which will enter a floating tank through an aperture in the bottom, which is 1½ feet square, the depth of the orifice below still water being originally 9 feet. Moreover, if the tank be 10 feet square in plan, and the top be originally

5 feet above water level, estimate how long it will take to sink the tank. You may assume that the coefficient of discharge is ·62, and that the weight per cubic foot of water is 64 lbs. (C. & G., H., Sec. C, 1904.)

6. Explain precisely the object and method of action of an air vessel when placed on the delivery pipe of a reciprocating pump, and sketch a self-acting arrangement whereby air may be injected into the air vessel to keep up the supply. Why does the efficiency of a reciprocating pump depend so much on the head against which it pumps? (C. & G., H., Sec. C, 1904.)

7. Answer *one* only of the following (*a*), (*b*), or (*c*) :—(*a*) Describe how you would proceed to determine experimentally (i.) the velocity ratio, (ii.) the mechanical efficiency of a differential pulley block. (*b*) Describe how you would carry out an experiment to determine the discharge of water through a sharp-edged circular orifice on the side of a water tank, say of half an inch diameter, its centre being two feet below the water surface. What sort of result would you expect to obtain? (*c*) Wishing to know the proper breadth and depth of a beam of some new kind of timber to carry a certain load, you experiment with a small beam, describe how you would proceed and what use you would make of your results. (B. of E., S. 2, 1904.)

8. In a hydraulic machine what resistance has the pressure to overcome? A load of 8 tons is raised by the ram of a hydraulic press for a distance of 15 feet. The head of water in the hydraulic mains supplying the ram with water is 1,600 feet. What must the volume of the ram be if the mechanical efficiency is 75 per cent.? (B. of E., S. 2, 1904.)

9. The ram of a hydraulic accumulator is 4 inches in diameter. The pressure of the water from the accumulator is to be $1\frac{1}{2}$ tons per square inch, and the water is used to work the hydraulic ram of a testing machine, which is $9\frac{1}{2}$ inches in diameter. (*a*) What total load must be placed on the 4-inch ram if 5 per cent. of the load is wasted in the friction of the cup leathers, &c.? (*b*) What total load is the testing machine capable of applying, if there is a further loss of 5 per cent. in the cup leathers of the large cylinder? (B. of E., S. 2, 1905.)

10. In a hydraulic crane the piston is 9 inches in diameter, and the velocity ratio (or the ratio between the velocity of the lift and the velocity of the piston) is 10. The water is delivered to the crane under a pressure of 1,200 lbs. per square inch, and the mechanical efficiency of the crane is 52 per cent. Find (i.) what load this crane will lift; (ii.) the quantity of water used in gallons per 35-foot lift. Why do these cranes have such a low mechanical efficiency? (B. of E., S. 2, 1905.)

11. In the case of a crane operated by hydraulic power explain, with the aid of a sketch, any device whereby severe shocks may be prevented if, when the load is being lowered, the exhaust valve is accidentally closed. (C. & G., H., Sec. C, 1905.)

12. Define the coefficients of "velocity," "contraction," and "discharge," as applied to the flow through orifices. The water in a cylindrical cistern of 1 square foot sectional area is 16 feet deep. Upon opening an orifice 1 square inch area in the bottom, the water fell 7 feet in one minute. What is the coefficient of discharge? (C. & G., H., Sec. C, 1905.)

13. A vertical wall, 12 feet high and 3 feet thick, built of masonry which weighs 150 lbs. per cubic foot, is subjected on one face to water pressure. What is the maximum depth of water behind the wall which is permissible, if the resultant force on the base of the wall is not to pass outside the outer edge of the wall. (B. of E., S. 2, 1907.)

14. The following results were obtained during an experiment to determine the quantity of water which would be discharged through a small

circular orifice in the side of a tank. The diameter of the orifice, which had sharp edges, was 1 inch :—

Number of Experiment.	Duration of Experiment.	Actual Discharge.	Head of Water above Centre of Orifice.
	Minutes.	Lbs.	Inches.
1	15	576	1·5
2	15	660	2·0
3	15	733	2·5
4	15	827	3·27
5	15	915	4·01
6	15	1,011	5·0
7	10	737	6·0
8	10	788	7·0

Plot on squared paper a curve to show the relation between the discharge in lbs. per minute and the head of water above the centre of the orifice. From your curve determine the discharge in gallons per hour when the head of water was 5½ inches. Plot also, on squared paper, a curve to show the relation between the discharge in lbs. per minute and the square root of the head of water above the centre of the orifice. From your curve what would you consider the relation to be between quantity of flow and head? Determine for each of the experiments in the above table the "coefficient of discharge" for this orifice, and plot a curve to show the relation between "coefficient of discharge" and head of water. (B. of E., S. 2 and 3, 1907.)

15. Make a section of a hydraulic riveter, pointing out the advantages of a hydraulic riveter, and sketch the accumulator which is most used with a riveter. (C. & G., 1907, H., Sec. C.)

16. Make a careful section of a double-acting plunger pump driven tandem from a steam engine. The valve passages and the arrangement of valves must be clearly shown. (C. & G., 1907, H., Sec. C.)

17. The sluice of a dock is closed by a vertical gate 6 feet wide and 5 feet deep; on the one side of the gate the head of water is 33 feet above the bottom edge of the gate, and on the other side the head of water is 15 feet above this edge. Find the resultant pressure upon this gate, and the point at which it acts. (B. of E., S. 2, 1908.)

18. Deduce an expression for the total pressure and the centre of pressure of water on a plane surface. A masonry dam, supporting water pressure, has a vertical face for 16 feet below the surface of the water, a face sloping 1 to 6 from 16 feet to 30 feet, and 1 to 3 from 30 feet to 42 feet. Find graphically the magnitude and position of the resultant water pressure for a length of 10 feet of the dam. (C. & G., 1908, H., Sec. C.)

LECTURE III.—ORDINARY QUESTIONS.

1. Water at a pressure of 700 lbs. per square inch is supplied to a hydraulic crane, and 11 cubic feet are used in lifting 15 tons through a height of 18 feet. How much energy has been given to the crane? How much energy has been wasted? (B. of E., Adv., 1903.)

2. Why are water-pressure engines generally uneconomical at low loads? Describe, with the aid of sketches, two methods by which a water-pressure

may be made economical at varying loads (i.) by a suitable arrangement of valves, (ii.) by varying the stroke. (C. & G., H., Sec. C, 1905.)

3. Sketch a hydraulic intensifier in sufficient detail to show the principle of its action, and give some illustration of its use.

(C. & G., H., Sec. C, 1906.)

Lecture IV.—Ordinary Questions.

1. A fire engine supplies water at a pressure of 40 lbs. per square inch to a pipe 3 inches diameter. The pipe is led a distance of 100 feet to a nozzle ¾ inch diameter, at a height of 25 feet above the pump. Find the quantity discharged per second and the height to which the jet can rise—having given that the coefficient of friction of the pipe is ·02, that the frictional resistances in the nozzle can be neglected, and that the actual height the jet rises is three-quarters of the head due to the velocity of efflux. Find also the pumping horse-power. (C. & G., H., Sec. C, 1903.)

2. Describe the two possible modes of flow of water along a pipe, and discuss any experiments that have been made to determine the velocity at which steady motion becomes eddy motion, or conversely. On what does the critical velocity depend; and how does the loss of head per unit length of pipe depend on the velocity of flow and the diameter of the pipe (i.) below the critical velocity, (ii.) above the critical velocity?

(C. & G., H., Sec. C, 1904.)

3. A hydraulic motor is fed through a 4-inch pipe from an accumulator one mile away, which works at a pressure of 700 lbs. per square inch. If the total loss of head in the mains is three times that due to skin friction alone, and if the coefficient of friction in the ordinary formula for loss of head is ·0075, estimate the horse-power delivered to the motor and the efficiency of transmission, when the velocity of the water in the pipe is 6 feet per second. (C. & G., H., Sec. C, 1904.)

4. Sketch the arrangement usually adopted in locomotives for picking up water without stopping. The height at which the water is delivered in the tender is 8 feet above the level of the water in the trough. Estimate the least speed of the locomotive, in miles per hour, at which the water may be picked up, assuming that half the kinetic energy of the water at the inlet is lost in overcoming frictional resistances. If, when the speed of the locomotive is 40 miles an hour, the sectional area of the stream at delivery into the tender is 10 inches by 5 inches, estimate the quantity scooped up in gallons per second, and also the additional resistance due to the action of the scoop. (C. & G., H., Sec. C, 1904.)

5. Distinguish between impulse and reaction turbines, and point out the relative advantages and disadvantages of each kind. An impulse turbine of the axial flow type receives water under a head of 50 feet in a direction making an angle of 30° with the tangent to the wheel. The water is discharged from the wheel in a direction parallel to the axis of rotation with a velocity due to a head of 5 feet of water. Neglecting frictional resistances of all kinds, find, graphically or otherwise, (i.) the circumferential velocity of the wheel, (ii.) the vane angle at inlet, and (iii.) the vane angle at exit.

(C. & G., H., Sec. C, 1904.)

6. Discuss, briefly, the advantages of compound centrifugal pumps when lifting against a big head. If a centrifugal pump is used simply to circulate water against a resistance—as in a condenser—and not to lift the water, show that, if the pump works under similar conditions at all speeds, the efficiency is constant, and the quantity of water circulated is proportional

to the speed of the pump—all hydraulic resistances, both in and outside the pump, being taken to vary as the square of the velocity of flow.
(C. & G., H., Sec. C, 1904.)

7. A hydraulic lift, with ram, load, &c., weighs 10 tons, the ram is 9 inches in diameter, and the friction in the mechanism is equal to 0·05 of the gross load. The accumulator is half a mile away and is loaded to 800 lbs. per square inch, the diameter of the supply pipe is 3 inches. Estimate the speed of ascent of the lift if the loss of head in the pipe is $0 \cdot 0005 l v^2/d$; l being length in feet, d diameter in feet, v the velocity of water in the pipe in feet per second. (B. of E., S. 3, 1904.)

8. A jet of water, of circular section, 1 inch in diameter, the velocity being horizontal and normal to the section, comes from a vessel which is at rest. The centre of the section is 5 feet below the atmospheric level of the water in the vessel. What is the velocity? What is the weight of water flowing per second? What is the momentum (in engineers' units) lost by the vessel per second? What is the horizontal force acting on the vessel?
(B. of E., S. 2, 1905.)

9. A set of pumping engines has to deliver 4,000,000 gallons of water per 24 hours through an 18-inch pipe, which is $7\frac{1}{2}$ miles long, and the total lift is 336 feet. If the coefficient of friction of the water in the pipe is $f = 0 \cdot 0055$. Find (i.) the additional lift due to the friction of the pipe; (ii.) the horse-power expended in maintaining this rate of discharge.

N.B.—The loss of head in feet due to friction when water is flowing through a pipe of diameter d feet and l feet long, with a velocity of V feet per second, is $\frac{V^3 l f}{2 g m}$. In this formula m is the hydraulic mean depth.
(B. of E., S. 3, 1905.)

10. Explain precisely why the discharge, under varying heads, of water over a triangular notch can be more accurately estimated than over a rectangular notch. How does the discharge over a triangular notch vary with the head, and why is the discharge over a broad rectangular notch very accurately expressed by the formula :—

$$\alpha (b - \beta h) h^{\frac{3}{2}},$$

in which b is the breadth of the notch, h the depth of water, and α and β two empirical coefficients? (C. & G., H., Sec. C, 1905.)

11. It is desired to pump 10 cubic feet of water per minute through a nozzle at the end of a hose pipe of 3 inches diameter and 100 yards long, and to deliver it at a height of 100 feet. The actual lift of the jet is three-quarters the head due to the velocity of efflux, and the combined efficiency of engine and pump is 0·6. Taking the ordinary law for the loss of head in the hose pipe, and assuming a coefficient of friction of 0·036, estimate the diameter of nozzle required and the I.H.P. of the engine.
(C. & G., H., Sec. C, 1905.)

12. Describe, with sketches, how you would measure the quantity of water flowing down a small stream. (B. of E., S. 2, 1906.)

13. In a shale mine in order to drain one of the pits a treble ram pump. driven by an electric motor, is employed. The rams are $9\frac{1}{2}$ inches diameter by 12-inch stroke, they each make 34·75 strokes per minute, the height to which the water is lifted is 393 feet, and the total length of the 6-inch discharge pipe is 700 feet. Find (*a*) how many gallons of water are lifted per minute; (*b*) how many foot-lbs. of useful work are done per minute; also, what is the useful horse-power when the pumps are running steadily; (c) how many foot-lbs. of work are done per minute in overcoming the friction in the pipe (the coefficient of friction is

0·0075) ; (*d*) the B.H.P. required to lift the water and overcome the pipe friction. (B. of E., S. 2 and 3, 1906.)

14. Show that the Lowell formula for a rectangular gauge notch is a rational formula. It is (for sharp edges at sides and sill)—

$$Q = 3{\cdot}33\left(L - \frac{1}{5}H\right)H^{\frac{3}{2}},$$

where Q is in cubic feet per second, L and H being in feet. Describe, with sketches, how you would measure the quantity of water flowing down a small stream. (B. of E., S. 3, 1906.)

15. State the usual formula for the loss of head in closed pipes, and point out how it may be experimentally determined in the case of small pipes, A horizontal lead pipe, 0·4 inch diameter and 3·48 feet long, has two pressure gauges inserted at the two ends. The difference of pressure for different rates of flow are observed, and the results are :—

Velocity of flow in feet per second (v),	8·04	11·67	14·43	17·41	19·90
Difference of head in feet of water (h),	3·03	6·11	9·07	12·21	15·62

Assuming $h \propto v^n$, find the best average value of n.
(C. & G., H., Sec. C, 1906.)

16. When 60 H.P. are delivered hydraulically into a mile of straight horizontal pipe at a pressure of 700 lbs. per square inch, it is found that 52 H.P. are available at the other end. If 40 H.P. are delivered at a pressure of 800 lbs. per square inch into a pipe having a diameter three-quarters that of the first, but double its length, estimate what horse-power would be available at the motor, the coefficient of friction being the same in both pipes. (C. & G., H., Sec. C., 1906.)

17. Explain the considerations which govern the discharge of water through orifices, and show that in no case can the coefficient of contraction be less than a half. (C. & G., H., Sec. C, 1906.)

18. A fire engine supplies water at a velocity of 6 feet per second in a pipe 3 inches diameter. This pipe is led a distance of 100 feet to a nozzle, $\frac{3}{4}$ inch diameter, 25 feet above the pump. If the coefficient of friction be taken as 0·02, and the actual lift three-quarters of the theoretical, find the height to which a vertical jet will rise, and the required pressure at the pump end of the main in lbs. per square inch. (C. & G., H., Sec. C, 1907.)

19. Describe some form of water meter for gauging the flow of water in pipes. (C. & G., H., Sec. C, 1907.)

20. A locomotive, going at 40 miles an hour, picks up water from a trough. The tank is 8 feet above the mouth of the scoop, and the cross-section of the delivery pipe is 50 square inches. If half the available head is wasted at entrance, find the velocity at which the water is delivered into the tank, and the number of tons lifted in a trench 500 yards long. What is the minimum speed at which the water can be picked up?
(C. & G., H., Sec. C, 1907.)

21. A direct-acting hydraulic lift is supplied with water from an accumulator at a pressure of 300 lbs. per square inch. The ram of the lift is 11 inches in diameter, the cylinder is 12 inches in diameter, and the supply pipe from the accumulator is 6 inches in diameter and 1,000 feet

long. What load can be raised at a steady speed of 3 feet per second if there is a loss of 20 per cent. due to the friction of the ram of the lift, and bends, &c., in the pipe? Use the formula for friction in the pipe, loss of head $= 0{\cdot}02\, l v^2/2\, g\, d$ (feet-second units). (B. of E., S. 3, 1908.)

22. The velocity of flow of water through a pipe delivering water into a house is 16 feet per second, and the length of the pipe is 60 feet. If a valve on this pipe is completely shut so that the quantity delivered per second diminishes at a uniform rate in 0·35 second, what will be the mean increase in pressure at a point near the valve? Neglect the motion of the water beyond the 60-foot length of pipe. (B. of E., S. 3, 1908.)

23. Water enters a vessel vertically and leaves by two horizontal jets each of 2 inches diameter with a velocity of 20 feet per second, one in the north and the other in the north-east direction; the vessel is kept at rest by a horizontal string, what is the pull in the string and what is its direction? Make a sketch giving your notion of how this experiment might be carried out. (B. of E., S. 3, 1908.)

24. The discharge of water from a reservoir is determined by the flow over a rectangular weir 8 feet wide with sharp edges to give full end contractions. The mean head over the weir is 9·3 inches; calculate the discharge in cubic feet per minute if the coefficient of discharge of the weir is 0·615. (C. & G., H., Sec. C, 1908.)

25. State the laws of friction for liquids against solid surfaces, and deduce an expression for the coefficient of friction for a liquid flowing through a pipe in terms of the dimensions of the pipe and the mean velocity of the water. In a laboratory experiment on a pipe $\frac{3}{8}$ inch internal diameter, and 72 inches in length between the gauge points, a discharge of 3,900 cubic inches was obtained in five minutes, and the mean difference of head between the gauge points was 8·54 feet of water. Assuming that the resistance to flow varies as the 1·73 power of the velocity, calculate the coefficient of friction. (C. & G., 1908, H., Sec. C.)

LECTURE V.—ORDINARY QUESTIONS.

1. The rim of an inward flow turbine moves at a speed of 30 feet per second, and the vanes are there at right angles to the rim. Water enters the rim with a radial velocity of 5 feet per second. If the water is to enter without shock, what must be the angle between the rim and the guide blades? Find the weight of water entering per second if the circumferential area of all the openings of the rim is 2·4 square feet. (B. of E., Adv., 1903.)

2. A centrifugal pump lifts 6·2 cubic feet of water per second to a height of 7 feet with an efficiency of 45 per cent. It is direct-driven by a continuous current electro-motor which works at 200 volts and whose efficiency is 85 per cent., what current must be supplied to the motor? The current comes from a station to the motor through leads whose resistance is 1·12 ohms; what horse-power is lost in heating the leads? (B. of E., S. 2, 1904.)

3. Why is the quantity of water actually lifted by a reciprocating pump invariably smaller than the theoretical quantity corresponding to the space swept through by the piston? Explain the advantages of having a number of small valves of low lift, rather than a larger valve of greater lift, and point out the effect on "slip" and "work expended" of having spring valves. (C. & G., H., Sec. C, 1905.)

4. Explain how a reaction turbine works, and how you would design the guides and vanes in order to ensure the greatest efficiency. What are the

advantages of "compounding" turbines? In a test of a small compound turbine the pressure at the entrance to the guides was 37 lbs. per square inch gauge, and it discharged into the atmosphere. When the revolutions were 1,644 per minute, the water used per minute was 366 lbs., and the turning moment on the spindle—as measured by a dynamometer—was 2·2 foot-lbs. Estimate the useful horse-power of the turbine and its efficiency. (C. & G., H., Sec. C, 1905.)

5. Make a sketch of a section of what you consider a well-designed centrifugal pump, and if the speed of the wheel at exit, the radial velocity with which the water leaves the wheel, and the vane angle at exit are all known, show how the work expended on the pump per lb. may be estimated. How does curving back the vanes at the discharging edge affect the efficiency, and what disadvantages arise if the angle is made too small? (C. & G., H., Sec. C, 1905.)

6. The supply of water for an inward-flow turbine installation is 500 cubic feet per minute, and the available head is 40 feet. The vanes are radial at the inlet, the outer radius is twice the inner, the constant velocity of flow is 4 feet per second, and the revolutions are 350 per minute. Find the velocity of the wheel, the guide and vane angles, the inner and outer diameters, and the width of the buckets at the inlet.
(C. & G., H., Sec. C, 1906.)

7. What is the object of adding a vortex chamber to a centrifugal pump, and why is a vortex chamber not so necessary when the blades are curved backwards? The velocity of flow through a centrifugal pump is 4·5 feet per second, and the blades make an angle of 20° with the tangent at the outer circumference. Find the velocity of the wheel rim so that the water may be pumped against an actual head of 6 feet, assuming the efficiency of the pump to be 0·6. (C. & G., H., Sec. C, 1906.)

8. The rim of a turbine is going at 50 feet per second. 100 lbs. of fluid enter the rim each second with a velocity in the direction of the rim's motion of 60 feet per second, leaving it with no velocity in the direction of the wheel's motion? What work is done per second upon the wheel? What work is done per lb. of fluid? If the total energy per lb. of fluid is h foot-lbs., and if 12 per cent. is lost in friction, find h. If the velocity of the water perpendicular to the rim's motion is 10 feet per second, and if the fluid enters the wheel without shock, draw the angles of the guide blade and vane at the entrance. (B. of E., S. 2 and 3, 1907.)

9. In an outward-flow turbine, with parallel crowns, the head is 140 feet, the discharge is 220 cubic feet per second, the flow velocity head is 8 per cent., and the other losses 14 per cent. of the available head. The velocity of the wheel at inlet is 0·9 the initial velocity of whirl. Design the guides and vanes, and determine the velocity of the wheel.
(C. & G., H., Sec. C, 1907.)

10. Make a section of a centrifugal pump suitable for pumping against a high head. Discuss what effect the shape of the casing has on the working of the pump and on the efficiency. (C. & G., H., Sec. C, 1907.)

11. In a test of a small 12-inch Pelton water-wheel the following results were obtained:—Mean effective load on rope brake, 27·2 lbs.; effective diameter of brake pulley, 7 inches; revolutions per minute, 308; diameter of nozzle of supply pipe, $\frac{1}{2}$ inch; quantity of water per hour passing through the wheel, 355·2 cubic feet. Determine (*a*) the horse-power given to the wheel by the water, (*b*) the horse-power given off at the brake, (*c*) the efficiency of the wheel. (B. of E., S. 2, 1908.)

12. Three cubic feet of water leave the wheel of a centrifugal pump per second; the radius there is 1 foot, breadth of the wheel is 0·1 foot, what

is the radial velocity of the water? The pump runs at 600 revolutions per minute, and the vanes direct the water backwards at 30° to the rim, what is the velocity of the water just after leaving the wheel? What is the component of this in the direction of motion of the rim of the wheel? (B. of E., S. 2, 1908.)

13. The pressure at the inside radius of the wheel of a radial-flow turbine is 2,000 lbs. per square foot; the wheel has a velocity of 12 radians per second; the inside radius is 0·4 foot, and the outside 0·8 foot. Very little water is flowing through, the head being nearly in a state of balance. Draw a curve showing how the pressure varies in the wheel. (B. of E., S. 2, 1908.)

14. Deduce an expression for the impulsive pressure of a jet of water impinging axially on a hemispherical cup, and show that the work done is a maximum when the velocity of the cup in the line of the jet is one-half the velocity of the water. Show what applications these results have had in the design of Pelton wheels, and also what modifications have been found desirable to secure high efficiency. (C. & G., H., Sec. C, 1908.)

15. Classify the methods of regulation of turbines, and state what are the advantages and disadvantages of each method. Describe one form of relay governor for a turbine, and point out any special features of interest in the form you describe. (C. & G., H., Sec. C, 1908.)

16. Describe one form of inward-flow reaction turbine. A turbine of this class is supplied with 600 cubic feet of water per minute at a head of 36 feet. The wheel is 24 inches in diameter at inlet and 12 inches in diameter at outlet. The peripheral speed of the wheel is 32 feet per second, and the radial velocity of the water through the wheel is 6 feet per second. Determine the breadths of the wheel and the blade angles at inlet and outlet if the water enters the wheel with a velocity due to 55 per cent. of the actual head and leaves radially. Also determine the angle of the guide blades at the periphery of the wheel and the hydraulic efficiency. (C. & G., H., Sec. C, 1908.)

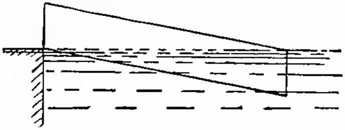
17. Explain what is meant by a forced vortex, and deduce an expression for the head produced by it in the wheel of a centrifugal pump between the inner and outer radii. Calculate the difference of head produced by the forced vortex set up in the wheel of a centrifugal pump running at 200 revolutions per minute if the inner diameter is 2 feet and the outer diameter 4 feet. (C. & G., H., Sec. C, 1908.)

Lecture I.—I.C.E. Questions.

1. A rectangular sluice, 4 feet square, turns on a horizontal hinge at its top edge, the hinges being immersed 2 feet. Find the total force on the sluice and also the horizontal force which must be applied at the bottom edge of the door in order to keep it closed against this pressure. (I.C.E., *Oct.*, 1903.)

2. Describe, with sketches, a hydraulic jack. The diameter of the ram of a hydraulic jack is 1¾ inch, the diameter of the plunger is ⅞ inch, and the mechanical advantage of the handle is 10. Find the total mechanical advantage. (I.C.E., *Oct.*, 1903.)

3. A uniform raft of rectangular section, which, when floating freely, is immersed to two-thirds its depth, has one end stranded so that the lower edge is in the plane of floatation, as in figure. If the ends be assumed vertical in the position of equilibrium, show that the upper edge of the sea end is also in the plane of floatation, and that the pressure on the wall end is W/4, Where W is the weight of the raft. (I.C.E., *Feb.*, 1904.)



4. A lock gate is 35 feet wide, and the heights of the water above the bottom of the gate on the two sides are 26 and 13 feet respectively. Find the resultant pressure and the height, measured from the bottom of the gate, at which it acts, the weight of water per cubic foot being 64 lbs. (I.C.E., *Feb.*, 1904.)

5. Describe the action of a bucket-and-plunger pump. What is meant by "slip" of a pump, and how can it be reduced? What are the other causes of loss of energy in such a pump? (I.C.E., *Feb.*, 1904.)

6. Give a sketch and explain the method of action of the pulsometer pump. (I.C.E., *Feb.*, 1904.)

7. A hydraulic crane is required to lift 4 tons through a height of 40 feet; the maximum travel of the ram is 8 feet, and the water pressure 700 lbs. per square inch. Determine a suitable diameter for the ram and sketch, and describe the arrangement of the cylinder and multiplying gear. (I.C.E., *Oct.*, 1904.)

8. Describe, with sketches, a hydraulic accumulator. The ram of an accumulator has a stroke of 20 feet, and the pressure in the cylinder is 1,000 lbs. per square inch. The capacity of the accumulator is 3 H.P.-hours. Find the diameter of the ram and the load on the carriage. (I.C.E. *Oct.*, 1904.)

9. A diving bell, of uniform horizontal section, 10 feet high, and which initially is full of air at atmospheric pressure, is sunk in water so that its top is 100 feet below the surface. If the barometric pressure be equivalent to 34 feet of water, find the height to which the air will ultimately rise inside the bell. (I.C.E., *Feb.*, 1905.

10. Define what is meant by the centre of pressure on a flat immersed surface. A tank, the plan of which is a square of 10 feet side, has a plane rectangular bottom inclined at 30° to the horizontal. If the minimum depth of water in the tank be 10 feet, estimate the total pressure on the

bottom of the tank and the depth of the centre of pressure below the surface of the water. (I.C.E., *Feb.*, 1905.)

11. A dock entrance is closed by a caisson, the depth of water being 24 feet over the level floor. The width of entrance is 40 feet at the floor and 48 feet at the water-level, the side walls having a straight batter. Find the total horizontal force due to hydrostatic pressure upon the caisson. Find the true centre of action of the horizontal pressure. (I.C.E., *Feb.*, 1905.)

12. Two pontoons are designed with the same breadth of 12 feet, and the same draught of 6 feet: the one having an immersed section of rectangular form 12 feet × 6 feet, while the immersed section of the other is a semicircle. Find the position of the metacentre in each case. (I.C.E., *Feb.*, 1905.)

13. Sketch carefully and describe some form of valve for controlling the supply and exhaust to a hydraulic crane or press. The packing arrangements are to be clearly shown. (I.C.E., *Feb.*, 1905.)

14. Make careful sketches showing in detail the construction of some form of double-acting, reciprocating pump, suitable for pumping water against a pressure of 700 lbs. per square inch or higher. The packing arrangements and the details of the valves are to be clearly shown. (I.C.E., *Feb.*, 1905.)

15. A ship sinks 2 inches on entering a river and then rises $1\frac{1}{2}$ inches on discharging 40 tons of cargo. Find its original displacement. Specific gravity of sea water = 1·025. (I.C.E., *Feb.*, 1906.)

16. Describe how water under pressure is caused to do work; and sketch a satisfactory form of hydraulic lift. (I.C.E., *Feb.*, 1906.)

17. State the law governing the transmission of pressure by fluids. In a hydraulic jack, the diameters of the ram and the plunger are 3 inches and 1 inch respectively, the lever arm of the force about the fulcrum is 20 times that of the plunger, and the efficiency is 70 per cent. Calculate the load lifted by a force of 50 lbs. What is the pressure in the hydraulic cylinder? (I.C.E., *Oct.*, 1906.)

18. A tidal flap, uniform in section and weighing 2,000 lbs., is 4 feet wide and 6 feet deep. It hangs on a horizontal hinge through the top edge, at an angle of 60° with the horizontal. Assuming that the water level on the outer side is 0·26 foot, measured vertically, below the hinge, find the water level on the inner side, relative to the hinge, when the flap is just about to open. (I.C.E., *Feb.*, 1907.)

19. A diving bell is in the form of a hollow circular cylinder 8 feet high inside, open at the bottom and closed at the top. When the axis is vertical and the bottom just touches the water surface, it is filled with air at atmospheric pressure, 15 lbs. per square inch, and at a temperature of 60° F. It is lowered gradually into the water, the axis remaining vertical, until its bottom end is 20 feet below the surface. Find the height to which the water rises inside the bell, and the pressure of the air inside, the temperature remaining the same. Find also the air pressure in lbs. per square inch which must be supplied to the bell to keep the water out. (I.C.E., *Oct.*, 1907.)

20. Find an expression for the depth below the surface of the centre of pressure on one side of a vertical plane immersed in a fluid. A circular plate is placed vertically in water, and just touches the surface. Its diameter is 3 feet. Find the depth of the centre of pressure on one side, having given that the second moment or moment of inertia of a circle about a diameter is $\frac{\pi}{64}d^4$. (I.C.E., *Oct.*, 1907.)

21. A U-tube of uniform section contains mercury. In one branch is also some water which fills 3 inches of the tube above the mercury in that branch. Find the difference in level of the surfaces of the mercury in the two branches, (*a*) when both ends of the tube are exposed to the atmosphere, (*b*) when the end containing mercury only is exposed to a pressure of 1 lb. per square inch, the other end being exposed to the atmosphere. The specific gravity of mercury is 13·6. (I.C.E., *Feb.*, 1908.)

22. Make sketches of *one* of the following:—(*a*) A hydraulic accumulator, (*b*) a hydraulic intensifier, (*c*) a hydraulic riveter with automatic return. The sketches should be in good proportion and show an accurate knowledge of detail. (I.C.E., *Feb.*, 1908.)

23. Explain why a pump cannot, by suction, lift water through a height equal to 34 feet. Why is it advisable to fit air vessels on to the suction and delivery pipes of reciprocating pumps, and why are such air vessels not necessary on the suction and delivery pipes of centrifugal pumps? (I.C.E., *Feb.*, 1908.)

24. A hydraulic accumulator has a lift of 5 feet; the diameter of the ram is 8 inches; the pressure per square inch in the accumulator is 700 lbs. per square inch; find the energy of the accumulator in horse-power hours. The accumulator works in conjunction with pumps working a hydraulic crane. The crane ram has a stroke of 6 feet and is 9 inches diameter; it makes a stroke in 15 seconds. If the crane ram makes a stroke every $1\frac{1}{2}$ minutes, find the time for each stroke of the crane during which the pump is idle, and the minimum horse-power of the pump. Compare it with the horse-power that would be required if there were no accumulator. (I.C.E., *Feb.*, 1908.)

25. The flow of water through a V notch is given by the formula $F = c\,H^{\frac{5}{2}}$, where c has the value 0·305 if F is in cubic feet per minute, and H the height of the water in the notch in inches. What are the dimensions of the coefficient c, and what is its value if F is in cubic metres per hour, and H is in centimetres? (I.C.E., *Oct.*, 1908.)

26. A vertical lock-gate has an aperture in it in the form of an equilateral triangle of 6 feet side, the lowest side being horizontal. The vertex of the triangle is 6 feet below the surface of the water. Find the total pressure and also the centre of pressure on a door closing the aperture? (I.C.E., *Oct.*, 1908.)

Lecture III.—I.C.E. Questions.

1. Explain by a sketch a method in which the pressure available in a hydraulic supply main may be intensified for use at a higher pressure. (I.C.E., *Feb.*, 1904.)

2. Describe, with the aid of sketches, a hydraulic intensifier. An intensifier is required to increase the pressure of 700 lbs. per square inch on the mains to 3,000 lbs. per square inch. The stroke of the intensifier is to be 4 feet, and its volume 3 gallons. Determine the diameter of the rams. (I.C.E., *Feb.*, 1905.)

3. Sketch and describe (*a*) some form of continuous registering meter, or (*b*) make diagrammatic sketches of a modern high-speed forging press, with intensifier and air accumulator. (I.C.E., *Feb.*, 1908.)

LECTURE IV.—I.C.E. QUESTIONS.

1. What quantities would it be necessary to measure in order to arrive at the total "head" of each lb. of water flowing in a pipe when it arrives at a given cross-section? Explain how they could be measured. (I.C.E., *Oct.*, 1903.)

2. In a divergent mouthpiece what considerations limit the size at the discharge end relatively to the contracted neck, in order that the mouthpiece may flow full? (I.C.E., *Oct.*, 1903.)

3. Water flows over a rectangular notch 3 feet wide to a depth of 6 inches, and afterwards passes through a triangular right-angled notch; find the depth of water through this notch. The coefficients of discharge for the notches are to be taken as 0·62 and 0·59 respectively. (I.C.E., *Oct.*, 1903.)

4. What is the formula for resistance to the flow of water in a pipe; if the diameter of a pipe is 2 feet 6 inches and the velocity of flow 3 feet per second, find the resistance to flow in a 100-foot length, and the work done per minute in overcoming this resistance. (I.C.E., *Oct.*, 1903.)

5. A semi-circular channel 10 feet in diameter flows full of water; compare its discharge with that of a rectangular channel of the same cross-sectional area 9 feet wide, lined with the same material, and having the same inclination. (I.C.E., *Oct.*, 1903.)

6. Two reservoirs, 10 miles apart, are connected by a pipe 3 feet in diameter, the difference in their water-levels being 40 feet; if the inlet valve to the lower reservoir is partially closed, so that the water rises in a vertical tube, let into the pipe on the inlet side of the valve 20 feet above the level of the water in the reservoir, what would be the discharge of the pipe? (I.C.E., *Oct.*, 1903.)

7. Water impinges on the bucket of a Pelton wheel, with a velocity of 100 feet per second, find the proper peripheral velocity of the wheel to obtain the maximum efficiency, giving proof. (I.C.E., *Oct.*, 1903.)

8. Explain the principle of the hydraulic ram, by means of which a stream of water can be made to pump part of its volume to a higher level. (I.C.E., *Oct.*, 1903.)

9. Water measuring 500,000 gallons per day is diverted from a stream at a point 30 feet above a hydraulic ram which it works; find what quantity of the water the ram can pump to a height of 200 feet above its own level. (I.C.E., *Oct.*, 1903.)

10. Draw a current-meter and describe fully how it can be used to find the discharge of a river. (I.C.E., *Oct.*, 1903.)

11. Write down a formula for the discharge of water over a weir and explain why it takes that particular form. How can the values of the coefficients be ascertained by experiment? (I.C.E., *Feb.*, 1904.)

12. At a drowned orifice 18 inches × 12 inches the water stands 12 inches higher on the approach side than on the downstream side; find the discharge through the orifice, and the width of a rectangular notch necessary to take this quantity of water with a depth of flow of 8 inches, the velocity of approach being negligible. (I.C.E., *Feb.*, 1904.)

13. A pipe 30 inches in diameter branches into two pipes of equal diameter whose combined area equals that of the 30-inch pipe; compare the loss of head in a mile of the latter pipe with that in a mile of the two pipes, the rate of flow being 4 feet per second. (I.C.E., *Feb.*, 1904.)

14. A channel, with a bottom width of 30 feet and side slopes of 2 : 1, flows full of water to a depth of 7 feet; find the velocity of flow in, and discharge of, the channel, the inclination being 1 in 5,000, and c in the Chezy formula 100. (I.C.E., *Feb.*, 1904.)

15. A right-angled elbow, 12 inches diameter, is bolted to the vertical side of a tank containing water, the free surface of which is 20 feet above the centre of the horizontal portion of the elbow. Determine the tension in the connecting bolts, due to the water pressure, when water is being discharged through the elbow under the head in the tank.
(I.C.E., *Oct.*, 1904.)

16. State Bernoulli's theorem. Water flows from an open tank into a vessel in which the pressure remains constant at 2 lbs. per square inch absolute, through a sharp-edged orifice 1 square inch in area. The surface of the water in the tank is kept at 15 feet above the centre of the orifice. Atmospheric pressure 15 lbs. per square inch. Find (i.) the velocity of discharge, (ii.) the discharge in gallons per hour. (I.C.E., *Oct.*, 1904.)

17. Five cubic feet of water are discharged per second from a stationary jet, the sectional area of which is $\frac{1}{2}$ square foot. The water impinges normally on a flat surface, moving in the same direction as the water, with a velocity of 2 feet per second. Find the pressure on the plane in lbs., and the work done on the plane in horse-power. (I.C.E., *Oct.*, 1904.)

18. A canal has a top width of 12 feet, bottom width 6 feet, side slopes 1 to 1, and longitudinal slope 2 feet per mile. Find the discharge per hour. State the value of the coefficient used, or use the value $c = 75$.
(I.C.E., *Oct.*, 1904.)

19. Describe, with the aid of sketches, the method you would adopt to gauge the flow of a small stream, and show how you would determine the quantity of water per hour. State any advantages or disadvantages of the method you adopt. (I.C.E., *Oct.*, 1904.)

20. A district of 10 square miles area drains into a large reservoir. The maximum rate at which the rain falls in the district is $1\frac{1}{2}$ inches in 24 hours. When rain falls after the reservoir is full it is discharged over a weir, the crest of which is level with the high-water level of the reservoir. Find the length of the weir so that the water in the reservoir shall never rise more than 15 inches above its high-water level. (I.C.E., *Oct.*, 1904.)

21. A pipe, 9 inches diameter and 1 mile long, connects two reservoirs. The pipe has a slope of 1 in 80. The level of the water is 25 feet above the inlet end and 6 feet above the outlet end. Neglecting all losses except skin friction, find the discharge and draw the hydraulic gradient. Determine the pressure head in the pipe at a distance of half a mile from the inlet. The coefficient of friction may be taken as 0·007. (I.C.E., *Oct.*, 1904.)

22. How many water mains, 6 inches in diameter, will be required to deliver 1,000 H.P. at a pressure of 700 lbs. per square inch. State carefully the velocity of flow assumed, and the reasons for choosing such a value. Calculate the percentage loss of energy due to friction for each mile length of the main. (I.C.E., *Feb.*, 1905.)

23. A conical pipe varying in diameter from 4 feet 6 inches at the large end to 2 feet at the small end forms part of a horizontal water main. The pressure-head at the large end is found to be 100 feet, and at the small end 96·5 feet. Find the discharge through the pipe. (I.C.E., *Feb.*, 1905.)

24. A pipe consists of half a mile of 12-inch and half a mile of 6-inch pipe, and slopes at 1 in 100. The discharge is 2 cubic feet per second. Find the difference in pressure-head at the two ends of the pipe.
(I.C.E., *Feb.*, 1905.)

25. A cast-iron sewer pipe is 15 inches diameter. It is laid at a slope of 1 in 1,000. How many gallons per day would be discharged—(*a*) when it is flowing full, (*b*) when the depth of the sewage is 6 inches?
(I.C.E., *Feb.*, 1905.)

26. A triangular notch, having an angle of 90 degrees, is used to measure the flow of a stream. Readings at intervals of 1 hour are taken, as shown in the table:—

Reading.	Head.	Reading.	Head.
1	4 inches.	4	7 inches.
2	5 "	5	6 "
3	6 "		

Draw a curve showing the rate of discharge at any time, and show how you would determine the discharge between the time of the first and last reading. (I.C.E., *Feb.*, 1905.)

27. Water under a head of 60 feet is discharged through a pipe 6 inches diameter and 150 feet long, and then through a nozzle, the area of which is one-tenth the area of the pipe. Neglecting all losses but the friction of the pipe, determine the pressure on a fixed plate placed in front of the nozzle. (I.C.E., *Feb.*, 1905.)

28. A small brass cylinder of uniform diameter, fitted with a non-return valve, is used as a depth-gauge in sounding operations. If after a sounding the top is unscrewed and the cylinder is found to be two-thirds full of water, what is the approximate depth it has reached? (I.C.E., *Oct.*, 1905.)

29. A line of piping has, in the upper portion of its length, a diameter of 15 inches for a length of 5,000 feet and an inclination of 4 per 1,000. A tapering pipe then reduces the diameter to 12 inches, which remains constant for a length of 2,000 feet, throughout which length the inclination is 3 per 1,000. Find the rate of discharge in cubic feet per second when the pipe is fully charged and is delivering freely at its termination. The equation of discharge may be assumed as $q = 42\sqrt{\frac{h d^5}{l}}$, where q denotes cubic feet per second, h the head lost in length l, and d the diameter in feet. (I.C.E., *Oct.*, 1905.)

30. Water is flowing through a circular pipe 4 feet in diameter under atmospheric pressure. At what depth is the discharge a maximum? (I.C.E., *Oct.*, 1905.)

31. A jet of water issues freely from a small orifice in the vertical side of a tank under constant head. The orifice is cut in a thin plate, and there is no loss of head at the orifice. Determine the path of the jet. (I.C.E., *Oct.*, 1905.)

32. Describe in algebraical terms the principle on which the Venturi meter is based. (I.C.E., *Oct.*, 1905.)

33. Describe the principle of action of (i.) a reducing valve, (ii.) an automatic ball air-valve, (iii.) a hydraulic ram for raising water. (I.C.E., *Oct.*, 1905.)

34. A jet of water, having an area of 6 square inches and a velocity of 50 feet per second, strikes perpendicularly a fixed plane of infinite extent. Find the pressure on the plane. Also find the normal pressure on the plane if the axis of the jet is inclined to the plane at an angle of 30°. (I.C.E., *Oct.*, 1905.)

35. A jet of water strikes a curved vane moving in a given direction with a given velocity. Show the form which must be given to the vane at the point at which the water strikes it in order that there may not be any shock. (I.C.E., *Oct.*, 1905.)

36. One hundred horse-power is to be transmitted to a distance of 5 miles with a loss of 15 per cent. of the head due to an accumulator pressure of 750 lbs. per square inch. The beginning and end of the pipe are at the same elevation. Find the diameter of the pipe. The equation of discharge

in Question 29 may be used in this case, but with a coefficient of 36 instead of 42. (I.C.E., *Oct.*, 1905.)

37. A pipe A B is fully charged with water at A. Two smaller pipes B C and B D convey the water from B to two points C and D. The length and diameter respectively of A B are 10,000 feet and 15 inches; of B C, 10,000 feet and 12 inches; of C D, 10,000 feet and 9 inches. Points C and D are respectively 50 feet and 80 feet below A. At all points the piping is under pressure except at C and D, where the water issues freely. Find the discharge at C and D, using the equation of discharge in Question 29. (I.C.E., *Oct.*, 1905.)

38. A tank 10 feet square and 10 feet deep has a circular orifice 4 inches in diameter in the bottom, which may be regarded as a thin plate. Water is admitted to the tank until it is full and is then shut off. In how many seconds will the tank be empty? (I.C.E., *Oct.*, 1905.)

39. Give the expression for the total energy of a pound of water flowing in a pipe; if the velocity of the water is 5 feet per second, its pressure 60 lbs. per square inch, and the height of the axis of the pipe at the point in question is 10 feet above the datum, find its value. (I.C.E., *Feb.*, 1906.)

40. A pipe 9 inches diameter gradually contracts to 4 inches diameter, its axis being level. The tops of the two legs of a U-tube containing mercury are connected by tubes to the centres of the two sections, and it is found that the difference in the level of the mercury in the two legs is 6 inches. What quantity of water is flowing through the pipe? The specific gravity of mercury is 13·6. (I.C.E., *Feb.*, 1906.)

41. Find what length of overflow weir is necessary to discharge a rainfall of 1 inch per hour over a watershed of 5,000 acres, if the height of the water over the crest of the weir is to be limited to 18 inches. (I.C.E., *Feb.*, 1906.)

42. Two service reservoirs of equal size, with vertical walls, are connected by a sluice 2 feet square. One of the reservoirs has been emptied for cleaning, and the depth of water in the other one is 20 feet. When the sluice is opened, find how long the water will take to attain the same level in the two if the area of each reservoir is 40,000 square feet. (I.C.E., *Feb.*, 1906.)

43. Derive the formula for the rate of flow of water in a pipe by equating the loss of head to the resistance. Taking a suitable value for the coefficient in the formula for velocity of flow, calculate the loss of head per mile in a pipe 30 inches diameter if the velocity of flow is 5 feet per second. (I.C.E., *Feb.*, 1906.)

44. Water flows out of a reservoir whose surface level is 150 feet above the datum for a distance of 5 miles, through a pipe 3 feet diameter, into a tank whose top water level is 125 feet above datum. Find the velocity of the water in the pipe and the rate of discharge into the tank. (I.C.E., *Feb.*, 1906.)

45. If, in the last question, a valve on the inlet to the lower tank be closed so as to reduce the velocity of flow to one-half of its original value, what would be the reading on a pressure gauge let into the pipe on the approach side of the valve? (I.C.E., *Feb.*, 1906.)

46. Find an expression for the power that can be transmitted by water flowing through a pipe of given size with a constant total fall. What proportion of the head would be lost in friction when the power delivered is a maximum? (I.C.E., *Feb.*, 1906.)

47. Define what is meant by "hydraulic mean depth," and find its value in the case of a channel with a bed-width of 30 feet and side slopes of $1\frac{1}{2}$ horizontal to 1 vertical. If the longitudinal slope is 1 in 6,000 and n in

Kutter's formula equals 0·02, find the rate of discharge of the channel. $c = \frac{\sqrt{m}}{n} \times \frac{x + 1{\cdot}811}{x + \sqrt{m}}$, when $x = n\left(41{\cdot}6 + \frac{0{\cdot}00281}{i}\right)$, m being the hydraulic mean depth and i the inclination. (I.C.E., *Feb.*, **1906.**)

48. What is meant by a "forced" vortex? If the wheel of a centrifugal pump is 9 inches inside and 18 inches outside diameter, and revolves at the rate of 1,500 revolutions per minute, to what height is it capable of raising water? Prove the formula made use of. If the pump is provided with a "whirlpool" chamber of twice the diameter of the wheel, to what additional height could the water be lifted? (I.C.E., *Feb.*, 1906.)

49. Define the principal causes of the variation of resistance to the flow of liquids in closed conduits and instance a formula comprising them. Give a general expression for each cause of resistance. Define (*a*) hydrostatic head; (*b*) frictional head; (*c*) hydraulic mean gradient; (*d*) hydraulic mean radius. (I.C.E., *Oct.*, 1906.)

50. Top water-level in a reservoir is 725 feet above datum. Give velocity in feet per second, and discharge in gallons per hour at a point X, 438 feet above datum and 11,976 yards from the reservoir, the main being 6 inches in diameter with a free outlet and an evaluated coefficient of 86. (I.C.E., *Oct.*, 1906.)

51. The discharge calculated in the preceding question decreases by 66 per cent. during the period of minimum flow. Give the pressure at X in lbs. per square inch during that period and the equivalent head in feet, the outlet being regulated to the reduced discharge. (I.C.E., *Oct.*, 1906.)

52. Show graphically the proportional dimensions and radii for egg-shaped sewers. Give the limiting velocities for sewers of all classes. State the rule for determining the size of an egg-shaped sewer from the ascertained diameter of a circular sewer of equivalent capacity. (I.C.E., *Oct.*, 1906.)

53. Give the diameter of a circular brick sewer to run half-full for a population of 80,000, the diurnal volume of sewage being 75 gallons per head, the period of maximum flow 6 hours, and the available fall 1 in 1,000. (I.C.E., *Oct.*, 1906.)

54. State the general formula for the discharge of water over a weir sharp-edged with a free overfall. Find the discharge in gallons per diem for a weir 3 feet wide with a still-water head of 2 inches. (I.C.E., *Oct.*, 1906.)

55. If the discharge last determined was passed through a right-angled triangular notch, what would be the head of water above the vertex of the notch? (I.C.E., *Oct.*, 1906.)

56. Sketch a submerged gauging-weir indicating the up-stream and down-stream heads. State how the discharge is calculated (*a*) by a single formula, and (*b*) by a double formula. (I.C.E., *Oct.*, 1906.)

57. Find the quantity of water discharged in gallons per minute from a circular orifice, 1 inch diameter, under a constant pressure of 34 lbs. per square inch, taking 0·62 as the coefficient of discharge. (I.C.E., *Feb.*, 1907.)

58. (*a*) Define the miner's inch; (*b*) express 10,000 gallons per hour in miner's inches. (I.C.E., *Feb.*, 1907.)

59. The discharge, stated in the previous question, when passed over a sharp-edged rectangular gauging-weir with two end contractions and a free overfall, produces a still-water head of $2\frac{1}{8}$ inches. Find the effective length of the weir. (I.C.E., *Feb.*, 1907.)

60. A rainfall of $\frac{1}{10}$ inch per hour is discharged from a catchment area of 5 square miles. Find the still-water head when this volume flows

over a weir with a free overfall, 30 feet in length, constructed in six bays, each 5 feet wide, taking 0·60 as the coefficient of discharge. (I.C.E., *Feb.*, 1907.)

61. Find the inclination necessary to produce a velocity of $4\frac{1}{2}$ feet per second in a steel water-main, 31 inches diameter, when running full and discharging with a free outlet, using the formula—

$$V = \sqrt[9]{\left(\frac{S \times 2g}{M}\right)^5 \times R^7},$$

in which M, the coefficient of rugosity, has a value of 0·0042. (I.C.E., *Feb.*, 1907.)

62. Calculate the quantity delivered by the water-main in the preceding question per day of 24 hours. This amount, representing the water supply of a city, is discharged into the sewers at the rate of one-half the total daily volume in 6 hours, and is then trebled by rainfall. Find the diameter of a circular brick outfall sewer which will carry off the combined flow when running half-full, the available fall being 1 in 1,500, and taking a value of 0·0054 for M in the formula given above. (I.C.E., *Feb.*, 1907.)

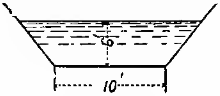
63. Define "momentum" and "moment of momentum." A jet of water issues at a velocity of 100 feet per second from a nozzle 1 inch in diameter. Find the reaction on the nozzle. (I.C.E., *Oct.*, 1907.)

64. A short horizontal pipe connects two vessels. The height of the surface of the water above the centre of the pipe, in one vessel, is maintained at 20 feet, and the vessel is open to the atmosphere. In the second vessel the surface of the water is 3 feet above the centre of the pipe and the pressure on the surface of the water is 3 lbs. per square inch absolute. Find the discharge between the two vessels, taking the coefficient of discharge of the pipe as 0·8. (I.C.E., *Oct.*, 1907.)

65. State a formula for the flow over a sharp-edged weir, explaining the meaning of each term. Describe where and how you would measure accurately the head over the weir, and discuss the effect of (*a*) the form of the approaching channel, (*b*) the velocity of the water in the channel, (*c*) the form of the sill of the weir—on the discharge. (I.C.E., *Oct.*, 1907.)

66. Show that in a channel of uniform section and slope, on the assumption that all the resistance is due to friction on the wetted surface of the channel, the resistance in lbs. per square foot of wetted surface is $F = w \, . \, m \, . \, i$, w being the weight of a cubic foot of water in lbs., m the hydraulic mean depth of the channel, and i the slope of the channel. Hence show, if F is supposed to vary with the square of the mean velocity v, that $v = c\sqrt{m \, . \, i}$. (I.C.E., *Oct.*, 1907.)

67. Find the discharge of a canal of the section shown by the diagram, having a slope of $2\frac{1}{2}$ feet per mile, and side slopes of 1 to 1. The



coefficient c in the formula given in the previous question may be taken as 95. (I.C.E., *Oct.*, 1907.)

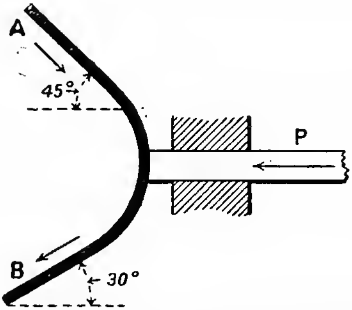
68. The rate of flow through a transverse section of a large river is required. Describe carefully with the aid of sketches how you would proceed to determine the flow, and the apparatus you would use. (I.C.E., *Oct.*, 1907.)

69. The difference of level of the surfaces of water in two reservoirs 8 miles apart is 60 feet. A pipe is required to convey 6,000 cubic feet per hour from the higher to the lower reservoir. Find the diameter of a cast-iron pipe such that the given quantity will flow along it when it has become dirty. At a point 3 miles from the upper reservoir the centre of the pipe is 100 feet below the surface of the water in the upper reservoir. Find the pressure in the pipe. (I.C.E., *Oct.*, 1907.)

70. State Professor James Thomson's principle of "Similarity" and use it to show that the discharge in cubic feet per second through a right-angled sharp-edged triangular notch is $Q = c \cdot H^{\frac{5}{2}}$, H being the still-water head over the apex of the notch in feet, and c a coefficient. If c is 2·635, find the discharge in gallons per hour when the still-water head over the apex of the notch is 4 inches. (I.C.E., *Oct.*, 1907.)

71. Find the horse-power of a pump required to raise 240 cubic feet per minute, through a height of 120 feet, the length of the delivery pipe being 2 miles and its diameter 1 foot, and the efficiency of the pump 0·6. The head lost due to friction in feet for this particular pipe may be taken as $h = 0{\cdot}0006\, v^2\, L$, L being the length of the pipe in feet and v the velocity of flow in feet per second. (I.C.E., *Oct.*, 1907.)

72. Two cubic feet of water per second strike a fixed cup, as shown in the diagram. The inclination of the stream to the horizontal when it strikes the cup is 45°, and when it leaves it the inclination to the horizontal is 30°, as shown in the figure. The velocity of the jet is 50 feet per second. If the cup is prevented from sliding horizontally by a force P, determine the magnitude of P in lbs. Find also the pressure on the cup perpendicular to P. (I.C.E., *Oct.*, 1907.)



73. State Bernoulli's theorem and show, by sketches, how it can be illustrated experimentally. Apply the theorem (a) to find the velocity of discharge of a fluid from a vessel through a small sharp-edged orifice at a given depth h below the free surface of the fluid in the vessel, (b) to find the discharge through a short horizontal conical pipe, having given the diameter at each end of the pipe and the pressure head at each end. (I.C.E., *Feb.*, 1908.)

74. State what is meant by the coefficients of velocity, contraction, and discharge of an orifice, and describe how you would determine them experimentally. A fluid, having a specific gravity of 0·5, is discharged from a vessel under a pressure of 50 lbs. per square inch absolute, through a sharp-edged vertical orifice 1 inch in diameter, into the atmosphere. Find the discharge in cubic feet per second. (I.C.E., *Feb.*, 1908.)

75. A jet, issuing from a fire-hose nozzle 1 inch diameter, rises to a height of 121 feet. Find the discharge of the nozzle, assuming the atmospheric resistance is zero. The pressure per square inch in the delivery pump is 100 lbs. per square inch. Determine the fraction of the head in the pump lost between the pump and the nozzle. Find also the force required to hold the nozzle in position. (I.C.E., *Feb.*, 1908.)

76. Determine the diameter of a brick-lined circular sewer, having a slope of 1 in 1,200, which will discharge 46 cubic feet per second when flowing full. The coefficient C in the formula $v = C\sqrt{mi}$ may be taken as

$$C = \frac{157}{1 + \dfrac{0 \cdot 3}{\sqrt{m}}},$$

or the logarithmic formula $i = \dfrac{0 \cdot 0001\, v^{1 \cdot 88}}{m^{1 \cdot 15}}$ may be used. (I.C.E., *Feb.*, 1908.

77. The heads lost by friction in two pipes, each 10,000 feet long, were as follows :—

Diameter of Pipe in Feet.	Velocity in Pipe in Feet per Second.	Head Lost in Friction.
d.	*v.*	*h.*
2	2	6·38
2	5	37·6
3	5	22·7

Show that the head lost by friction can be expressed by the formula

$$h = \frac{0 \cdot 0004\, v^{1 \cdot 93}\, l}{d^{1 \cdot 25}},$$

l being the length of the pipe in feet. (I.C.E., *Feb.*, 1908.)

78. Find the time taken for the water in a tank, 80 feet long by 20 feet wide, to fall 8 feet when discharging through a horizontal pipe, 12 inches diameter and 400 feet long, having a sharp edge at both ends, and the centre of which is 16 feet below the surface of the water when the flow commences. (I.C.E., *Feb.*, 1908.)

79. Explain how the principle of the conservation of energy is applied to obtain an expression for the energy of water due to its elevation, pressure and velocity. A water-main discharging 300 gallons per minute has a diameter of 5 inches at a gauge point where the pressure is 45 lbs. per square inch. At a second gauge point 25 feet below the first, and with a reduced diameter of 4 inches, the pressure in the pipe is 40 lbs. per square inch. Calculate the loss of head between the two sections due to friction and other resistances. (I.C.E., *Oct.*, 1908.)

80. A miner's inch has been defined to be the discharge through an orifice in a vertical plane of 1 square inch in area under an average head of 6·5 inches of water. Show in what ways this definition lacks precision, and what further qualifications are necessary in order that the discharge shall be perfectly defined. Calculate the discharge per second from a circular orifice 1 inch in diameter under a head of 2 feet, if the coefficient of discharge is 0·62. (I.C.E., *Oct.*, 1908.)

81. Describe the construction and action of a Venturi meter, and deduce a formula for the discharge from a meter of this type. A Venturi meter in a 30-inch pipe line has a throat diameter of 12 inches, and the mean difference of head between the large and small sections was found to be 14·6 feet of water during a period of 24 hours. Calculate the average discharge from the meter per hour. (I.C.E., *Oct.*, 1908.)

82. Assuming the formula $v = c\sqrt{mi}$ show that the discharge of a channel of rectangular cross-section varies as

$$\frac{d^{\frac{3}{2}}}{\sqrt{b + 2d}},$$

where b is the breadth of the channel and d is the depth of water in it. A rectangular channel 40 feet wide is found to discharge 240 cubic feet per second when the depth of water is 2·5 feet. Calculate the discharge per second when the depth is 2·73 feet. (I.C.E., *Oct.*, 1908.)

83. Show that, on the assumption that the relation between the head h and the velocity v of water flowing over a V notch is expressed by $v^2 = 2gh$, the discharge, Q, can be expressed in the form

$$Q = \frac{4}{15} c B H \sqrt{2gH},$$

where c is an experimental coefficient, B is the breadth of water at the weir and H is the head measured from the apex of the notch. Calculate the discharge per minute over a right-angled notch if the head is 9·3 inches and the value of the coefficient is 0·617. (I.C.E., *Oct.*, 1908.)

84. State the laws of friction of water against solid surfaces, and express these laws in a form applicable to the case of the loss of head due to the friction of water in a pipe. The following observations were recorded of an experiment on a 12-inch water-main having a distance between the gauge points of 100 feet:—

Discharge per Minute in Cubic Feet.	Loss of Head in Feet of Water.	Discharge per Minute in Cubic Feet.	Loss of Head in Feet of Water.
61·3	0·053	179·1	0·352
103·7	0·136	230·9	0·545
136·7	0·212	273·3	0·740

Determine the relation between the loss of head h and the mean velocity of the water v, assuming that it has the form

$$h = \frac{c v^n l}{d^{\frac{5}{4}}},$$

where l is the length and d is the diameter of pipe and c is a coefficient. (I.C.E., *Oct.*, 1908.)

85. Describe how you would proceed to gauge the discharge of a river about 300 feet wide and having a maximum depth of 35 feet. Explain in some detail the apparatus you would prefer to use, the observations you would take, and how you would reduce them to obtain an accurate value of the discharge. (I.C.E., *Oct.*, 1908.)

Lecture V.—I.C.E. Questions.

1. The casing of a centrifugal pump is full of water, but the valve in the delivery pipe is closed; if the diameter of the pump-disc is 12 inches, and it is rotated at 1,200 revolutions per minute, find the pressure at the delivery valve, the length of the suction-pipe being negligible. (I.C.E., *Oct.*, 1903.)

2. Explain the essential differences in working and design of impulse and pressure or reaction turbines. (I.C.E., *Feb.*, 1904.)

3. In an inward-flow turbine, subject to a constant head of water, why is the distribution of pressure in the wheel conducive to steady running when the power required varies, and why does the opposite effect obtain in the case of an outward-flow turbine? (I.C.E., *Feb.*, 1904.)

4. What is meant by the efficiency of a turbine? Explain fully how you would proceed experimentally to find the rate of running that would give the best efficiency for a particular turbine. (I.C.E., *Feb.*, 1904.)

5. Sketch and explain the construction of a centrifugal pump, and point out the means taken to diminish the losses of energy. (I.C.E., *Oct.*, 1904.)

6. An inward-flow turbine wheel works under a head of 60 feet, and makes 380 revolutions per minute. The diameter of the outer circumference of the wheel is 24 inches and of the inner circumference 12 inches. The velocity of the water entering the wheel is 44 feet per second, and the angle it makes with tangent to the wheel is 10 degrees. Assuming the velocity of flow through the wheel to be constant, determine graphically the direction of the tangent to the vane of the wheel at the inlet and outlet. Sketch a suitable form for the vane. Determine the hydraulic efficiency of the turbine. (I.C.E., *Oct.*, 1904.)

7. Show that the efficiency of a Pelton wheel is a maximum—neglecting frictional and other losses—when the velocity of the cups equals half the velocity of the jet. 25 cubic feet of water are supplied per second to a Pelton wheel through a nozzle, the area of which is 44 square inches. The velocity of the cups is 41 feet per second. Determine the H.P. of the wheel, taking a reasonable value for the efficiency. (I.C.E., *Feb.*, 1905.)

8. Prove, that if a mass of water enter the wheel of a centrifugal pump with no component tangential to the wheel, the theoretical height to which the water will be lifted is $h = \frac{V v}{g}$, where V equals the velocity of the outer periphery, and v the component of the velocity of the water when it leaves the wheel along the tangent to the wheel. If V is 60 feet per second, and the radial velocity of flow 5 feet per second, and the angle the tangent to the vane makes with the direction of motion is 120 degrees, determine h, and also the pressure-head at the point where the water leaves the wheel. (I.C.E., *Feb.*, 1905.)

9. AB and AC are two lines inclined at 30 degrees. A jet of water moves in the direction AC with a velocity of 24 feet per second, and a vane in the direction AB with a velocity of 12 feet per second. Show how to find the form of a vane so that the water may come on it tangentially, and leave it in a direction perpendicular to the direction of motion of the vane. Determine the pressure on the vane in the direction of motion due to each pound of water striking the vane. (I.C.E., *Feb.*, 1905.)

10. A reaction wheel is revolving with a given velocity V, and from its orifices water is issuing with a given velocity v. Show that for each lb. of water the useful work $= \frac{V(v - V)}{g}$, and the efficiency $= \frac{2V}{v + V}$. (I.C.E., *Oct.*, 1905.)

11. A Pelton wheel 3 feet diameter revolves at the rate of 200 revolutions per minute, the velocity of the jet of water impinging on the buckets being 80 feet per second. Find the hydraulic efficiency of the wheel. What should its velocity be for the efficiency to be a maximum ? (I.C.E., *Feb.*, 1906.)

12. Define impulse and reaction turbines. What is the efficiency of a turbine delivering 5,000 horse-power when 193,620 gallons of water per minute pass through it under a head of 136 feet? (I.C.E., *Oct.*, 1906.)

13. Water flows through a circular penstock, 30 inches in diameter, at a mean velocity of 50 feet per minute. Find the head of water at the centre of the opening when the discharge is 450 gallons per minute. (I.C.E., *Feb.*, 1907.)

14. (*a*) Sketch an end view or vertical section of an overshot water-wheel, indicating the head and tail races, total head, entrance fall, exit fall, and the angles of entry and exit. (*b*) State the two fundamental requirements necessary to secure high efficiency. (I.C.E., *Feb.*, 1907.)

15. (*a*) Find the theoretical H.P. of an overshot water-wheel, 22 feet diameter, using 2,000,000 gallons of water per 24 hours under a total head of 25 feet. (*b*) Express the hydraulic efficiency of an overshot wheel in terms of head, entrance fall, and exit fall. (I.C.E., *Feb.*, 1907.)

16. Make sketches showing the construction of a centrifugal pump. Explain the principal causes of loss of energy in the centrifugal pump, and show how in practice attempts are made to diminish these losses. (I.C.E., *Oct.*, 1907.)

17. A quantity of water, Q cubic feet per second, flows through a turbine, the initial and final directions of flow being known. Find the couple exerted on the turbine, and hence, when the angular velocity of the turbine is known, the work done on the turbine. If h is the total difference of level between the head and tail water of the turbine, u the velocity with which the water leaves the turbine, and the hydraulic efficiency of the turbine is η, show that the total fraction of the head lost in the supply pipe, turbine wheel and casing, and delivery pipe, is

$$k = 1 - \eta - \frac{u^2}{2gh}.$$ (I.C.E., *Oct.*, 1907.)

18. The water approaching a water-fall having a drop of 100 feet is measured by a weir 20 feet long ; the head over the weir is observed to be 9 inches. If the water, instead of being allowed to pass over the fall, is led into a penstock and used to drive turbines, and 65 per cent. of the available energy is converted into useful work, find the horse-power of the turbines. The coefficient for the weir is to be taken as 3·33. (I.C.E., *Feb.*, 1908.)

19. It is desirable that the turbines of Question 6 should be placed at least 15 feet above the bottom of the fall. Show, by sketches, the kind of turbine you would use, explaining carefully how full advantage is taken of the available head. If the velocity with which the water from the turbine enters the tail race is 10 feet per second and other losses are neglected, find the efficiency of the turbine. (I.C.E., *Feb.*, 1908.)

20. Describe, by aid of sketches, either (i.) the principal features of a hydraulic power-station for supplying water at 700 lbs. pressure per square inch to a large city, or (ii.) a hydraulic installation for utilising by means of turbines the natural fall of a river of about 40 feet. (I.C.E., *Oct.*, 1908.)

21. A jet of water 2 square inches in area and at a velocity of V feet per second impinges upon a set of hemispherical vanes, and drives them in its

own direction at a velocity of v feet per second. Deduce an expression for the work done, and plot a curve showing its amount for all possible speeds of the vanes when V is 400 feet per second. Describe a modification of the hemispherical vane used in a Pelton wheel, and show what are the practical reasons for the departure from the hemispherical form.

(I.C.E., *Oct.*, 1908.)

22. Describe by aid of sketches the construction of one form of rotary hydraulic engine operated by water at a high pressure, and explain what circumstances limit the speed of such an engine, and what causes may lead to a reversal of stress in the engine. (I.C.E., *Oct.*, 1908.)

23. Describe the principal considerations which you would take into account in the design of a turbine suitable for a low fall liable to fluctuations of level and water-supply. Describe the construction and special features of advantage of a form of turbine which you consider particularly well adapted to give a high all-round efficiency with a water-supply which is normally 6,000 cubic feet per minute at 10 feet head rising to 9,000 cubic feet per minute at a head of 6 feet in times of flood. (I.C.E., *Oct.*, 1908.)

24. Describe and illustrate, by sketches in good proportion and detail, the construction and action of a high lift centrifugal pump capable of lifting water against a head of not less than 300 feet. Explain what are the essential features of difference between the form you describe and that of a low lift centrifugal pump designed for a head of 30 feet.

(I.C.E., *Oct.*, 1908.)

APPENDIX B.—(*Continued.*)

ANSWERS TO QUESTIONS AT END OF LECTURES.

LECTURE I.—INST.C.E.

1. Specific gravity of the liquid = 1·25.
3. Resultant pressure on surface of body = 147 lbs.
5. Total pressure on area = 1,320 tons.
8. Total pressure = 698¾ lbs. Vertical component = 280·8 lbs. Pressure on base = 249·6 lbs. Weight of wedge = 31 lbs.
11. Height of water = 6·7 feet.
18. Work done per stroke = 290·5 foot-lbs.
19. The maximum height of the column of water is 33·7 feet.
23. P = maximum stress on inside face of cylinder. R = outer radius. p = pressure per square inch of cylinder. r = inner radius per inch. $P = p\frac{R^2 + r^2}{R^2 - r^2}$.
25. Pressure on top screw = ·65 lb. Pressure on bottom screw = 3·25 lbs.
28. 6·98 H.P.
30. Force required to open valve = 1,337 lbs.
33. Pressure on each gate = 7·3 tons, and pressure between the two gates = 13 tons.

LECTURE III.—ORDINARY.

6. 7½ inches for differential accumulator. 2·22 inches for solid ram.
8. 20·8 inches diameter.

LECTURE III.—INST.C.E.

1. Energy stored = 100,800 foot-lbs.
3. 13·29 gals. Load on press = 19·15 tons. Energy stored = 191·5 ft.-tons.

LECTURE IV.—ORDINARY.

7. 184·5 B.H.P.
20. 207,000 gallons, and 1,778 H.P.
22. H.P. lost, 3·85.
23. Discharge = 563·75 gallons per hour.

LECTURE IV.—INST.C.E.

3. Q = 176 cubic feet per second.
4. 1·1 foot per second, taking $f = {\cdot}01$; Q = ·867 cubic foot per second.
7. Pressure = 618·3 lbs.
8. Force on cup = 221 lbs.
9. 91·2 cubic feet per hour, with the coefficient $c = {\cdot}62$.
10. Time of emptying the lock $= \dfrac{2\,\Omega}{c\,\omega}\sqrt{\dfrac{H}{2g}}$.
12. L = 3·6 feet, taking the coefficient $c = {\cdot}7$.
13. h_L = 8·43 feet per mile.
15. Slope = ·4 foot per mile.
16. (1) For new pipes, $f = {\cdot}005\,(1 + \frac{1}{12}d)$; (2) and (3) $f = {\cdot}01\,(1 + \frac{1}{12}d)$.

$$v^2 = \frac{2g}{f} \times \frac{A}{P} \times \frac{h}{l}.$$

17. Delivery = 14·1 cubic feet per second.
19. $h_L = \left\{(m-1)^2\right\}\dfrac{V^2}{2g} = \left\{\left(1 - \dfrac{1}{m}\right)^2\right\}\dfrac{V^2}{2g}$.

20. Kempe's formula:—Diameter of delivery pipe $= \sqrt[6]{\dfrac{4\,q^2\,(l+5\,d)}{h}}$; diameter of supply pipe $= \sqrt[5]{\dfrac{2\,Q^2\,(L+5\,D)}{H}}$. Delivery in cubic feet per second $= \dfrac{\eta\,Q\,H}{h}$, where η is the efficiency of the ram.

21.

Kinetic Energy.	Preserved Energy.	Potential Energy.
Turbines. Pelton wheels.	Accumulators. Capstans. Hydraulic lifts.	Water wheels of all kinds.

22. $Q = a\sqrt{\dfrac{2g\,(h_1 - h_2)}{1 + 4f\dfrac{l}{d}}}$.
23. Difference in pressure = $\cdot 67\,v^2$.
25. Depth of water over weir = 11·46 inches.
26. Time, 3 hours 7 minutes.
28. Depth of water in pipe = $\cdot 95\,d$.
30. 2 : 2·8 H M D. $\dfrac{v_1}{v_2} = {\cdot}913\sqrt{\dfrac{h_1}{h_2}}$.
31. $\dfrac{v_1}{v_2} = \dfrac{1}{2}$.
33. Q = 1,040,000 gallons per day.
34. v = 1·81 feet per second; Q = 65·16 cubic feet per second.
35. (*a*) The coefficient of discharge will increase slightly, and therefore the discharge will increase; (*b*) depth and width of square orifice = ·2383

foot, allowing a coefficient of discharge of ·62; (*c*) the coefficient of discharge increases as the ratio of width to height increases, hence the discharge will be greater and in the ratio of about 1·05 to 1.

36. (A.) No. 1 lets the greater quantity pass by 12 per cent.; (B.) No. 1 lets the greater quantity pass by 11·5 per cent. (B.) No. 1, $v = 6·86$ feet per second; No. 2, $v = 6·59$ feet per second; Mean $v = 6·725$ feet per second.

40. (*a*) 41·6 lbs.; (*b*) 16·3 lbs. Work done = 666·6 foot-lbs.

41. Pressure = 498·37 lbs. 42. 10 H.P.; loss ·11 per cent.

43. ·46 per cent. H.P. is lost; for each pair of flanges add 1 foot to length of pipe; now, taking 6 pipes per 100 feet, there is a saving in weight of 47 lbs. per 100 feet.

45.

Velocities in Feet per Second.	Percentage Loss.		
	$p = 200.$	$p = 700.$	$p = 1,200.$
$v = 2$	$3\frac{1}{3}$	·95	5·55
$v = 3$	$7\frac{1}{2}$	2·1	12·49
$v = 4$	$13\frac{1}{3}$	3·8	22·2

46. $Q = c\,\frac{4}{15}\,b\,h\,\sqrt{2gh} = 594$ gallons per minute, where $c = 0·6$.

47. Full, 1 foot; half-full, 1 foot; 2·193 feet. Velocities, taking $c = 140$, are 2·5 and 3·78 feet per second.

50. $v = 10·9$ feet per second; $Q = ·009$ cubic feet per second; $E_K = 1·83$ foot-lbs. per second.

51. $d = 2·4$ inches. 52. Loss of head = 41·25 feet.

53. Q = 23·5 cub. feet per second. 55. F = 542·5 lbs. per square inch.

57. 4·34 feet per second; 119,700 gallons per hour.

58. $h_L = ·92$ feet.

60. Total pressure = $55\frac{1}{4}$ lbs.; max. pressure = 3·9 lbs. per square inch.

63. Reaction = 153·4 lbs. 64. 1·22 lbs. per square inch.

65. $c = 88$; $v = ·067$ foot per second; Q = ·67 cubic feet per second.

66. Q = 4·85 cubic feet per second.

68. Percentage loss of head = 32·5; H.P. = $96\frac{3}{7}$.

LECTURE V.—INST.C.E.

4. Angle $\theta = 49°$ at outer circumference. $\beta = 12°$ at inner circumference.

8. Angle of blade = 30°. 10. H.P. transmitted = ·0064.

11. Total pressure = 45,550 lbs.

19. Angle at entrance = 73° 41′ to horizontal. Angle at exit = 106° 19′ to horizontal.

21. Tangent at inlet, 60° 30′ to radius; tangent at outlet, 50° to radius.

23. Maximum efficiency = 75 per cent.

ANSWERS TO QUESTIONS IN APPENDIX B.

Lecture I.—Inst.C.E.

1. Total force on sluice = 3,994 lbs.; horizontal force applied = 2,330 lbs
2. Total mechanical advantage is 40.
4. Resultant pressure = 253·6 tons, and the height at which it acts from the bottom of the gate = $10\frac{1}{9}$ feet.
7. Diameter of ram = 9 inches.
8. Diameter of ram = $19\frac{1}{2}$ inches; load on the carriage = 132·6 tons.
9. Height to which the air will rise inside the ball = 7·5 feet.

Lecture III.—Inst.C.E.

2. Diameter of the ram = 4·7 inches; diameter of piston = 9·72 inches.

Lecture IV.—Inst.C.E.

3. Depth of water through notch = 13·7 inches.
4. $h_L = f\left(\frac{4lv^2}{d2g}\right)$. Resistance to flow, h_L = ·169 foot; and work done per minute in overcoming this resistance = ·282 H.P.
5. Ratio of discharge, 1·065 : 1.
6. Discharge of pipe = 11 cubic feet per second.
7. Peripheral velocity of Pelton wheel = 50 feet per second.
9. Quantity of water pumped by hydraulic ram, allowing an efficiency of 60 per cent., = 45,000 gallons.
12. Discharge from orifice = 7·44 cubic feet per second; width of rectangular notch = 4·13 feet.
13. The loss of head in each case is 15·84 feet and 44·8 feet.
14. The velocity of flow of the channel = 3·17 feet per second; discharge of channel = 977 cubic feet per second.
15. Tension in the connecting bolts = 1,226 lbs.
16. (i.) Velocity of discharge = 53·8 × ·97 = 52·1 feet per second; (ii.) discharge = 5,200 gallons per hour.
17. Pressure on the plane = 62 lbs.; work done per minute on the plane = ·225 H.P.
18. Discharge = 193,700 cubic feet per hour.

20. Length of weir = 87·3 feet, when using a coefficient of discharge of ·62.

21. Discharge of water from pipe = 2·3 cubic feet per second; hydraulic gradient is 1 in 62·4; pressure head in the pipe at a distance of half a mile from the inlet = 15·3 feet.

22. Assumed velocity of flow = 3 feet per second; number of water mains required = 10; percentage loss of energy for each mile length of the main due to friction = 2·75 per cent.

23. Velocity of flow = 15·32 feet per second; discharge through the pipe = 48·16 cubic feet per second.

24. Difference in pressure head at the two ends of the pipe = 211·8 feet.

25. Discharge (*a*) 1,067,000 gallons per day; (*b*) 388,000 gallons per day.

27. Pressure on fixed plate placed in front of nozzle = 135 lbs.

29. Rate of discharge from pipe = 3·55 cubic feet per second.

30. Depth of maximum discharge of water from a circular pipe = 3·8 feet.

31. The path of the jet is a parabola whose equation is $y = \frac{g x^2}{4 g h}$, where h is the head.

34. Pressure on the plane = 203 lbs.; normal pressure on the plane, when the axis of the jet is inclined at an angle of 30° to the plane, = 27·2 lbs.

36. Diameter of pipe = 5·88 inches.

37. Discharge at C and D = 1·82 and 1·43 cubic feet per second respectively.

38. Time taken to empty the tank = 24·35 minutes, when using a coefficient of discharge of ·62.

LECTURE V.—INST.C.E.

1. Pressure at delivery valve = 17·7 lbs. per square inch.

6. Direction of tangent to the vane of wheel at inlet = 113° 6′; direction of tangent to the vane of wheel at outlet = 20° 54′; hydraulic efficiency of turbine = $66\frac{2}{3}$ per cent.

7. H.P. of Pelton wheel = 235·8 H.P., taking an efficiency of 80 per cent.

8. h = 106·4 feet; the pressure head at point where the water leaves the wheel = 55·4 feet.

9. The vane makes at inlet an angle 126° 12′ with AB; the vane makes at outlet an angle 45° with AB; pressure on vane in the direction of motion due to each 1 lb. of water striking it = ·275 lb.

APPENDIX C.

THE INSTITUTION OF CIVIL ENGINEERS' EXAMINATION, FEBRUARY, 1909.

ELECTION OF ASSOCIATE MEMBERS.

GROUP II.—HYDRAULICS.

Not more than EIGHT *questions to be attempted by any Candidate.*

1. Show how to determine the resultant pressure on a curved surface subjected to a uniform hydrostatic pressure. A 12-inch water main has a bend subtending an angle of 60° at its centre of curvature. The water in the pipe is under a mean head of 65 feet. Determine the magnitude and position of the force tending to displace the bend.

2. An area of 5 square miles forms the catchment basin for a reservoir, and the maximum rainfall is 1·75 inches in 24 hours. In times of flood the water is diverted by a separating weir 36 feet long. Describe the construction of one form of weir suitable for this purpose, and calculate the average height of the water over the weir crest, assuming that the maximum rainfall for one day will pass over the weir in 36 hours, and that the coefficient of the weir is 0·7.

3. Obtain a general expression for the flow through a submerged orifice and apply it to determine the time required to empty a lock, of 15,000 square feet of superficial area and having vertical walls, if the initial difference of level between the lock and the lower reach of the canal is 8 feet, the area of the sluice openings is 30 square feet, and the coefficient of discharge is 0·64. The level of the lower reach of the canal is assumed not to be influenced by the discharge.

4. Describe by aid of sketches the construction of one form of meter suitable for measuring small quantities of water supplied at irregular intervals, such as occurs in a house supply. Explain what are the special features of advantage of the form you describe.

5. Explain what is meant by (1) a free spiral vortex, (2) a forced vortex, and give two examples of each kind of vortex motion which occur in engineering practice. The wheel of a centrifugal pump has an external diameter of 3 feet, and an internal diameter of 1·5 feet. The pump casting is filled with water, and with the discharge pipe closed, the wheel is revolved at the rate of 300 revolutions per minute. Calculate the difference of head set up between the inner and outer rims of the wheel

6. Explain concisely the special features of difficulty which arise in regulating the speed of a turbine, and describe by aid of sketches the

essential features of one form of governing gear suitable for a turbine of about 1,000 horse-power.

7. Describe the principal features of construction of rectangular measuring weirs ordinarily used for waterworks purposes, and the general conditions to be observed in order to obtain an accurate measure of the flow. Deduce a formula for the flow from such a weir in terms of its dimensions and the head.

8. Explain under what circumstances it is useful to plot the logarithmic values of the results obtained from hydraulic measurements, and quote some examples where the method is of value. In a series of measurements of the discharge from a hydraulic machine, the following values were obtained of the relation of the head of water to the discharge as follows :—

Discharge (Q) in Cubic Feet per Minute.	Head (H) in Feet.
2·56	0·192
7·95	0·302
23·10	0·463
83·52	0·774

It is known that the relation between Q and H is of the form $Q = CH^n$. Determine the values of c and n from the measurements, and calculate the discharge for $H = 0·623$ foot.

9. Describe the construction and action of one form of hydraulic accumulator. Water at a pressure of 2,000 lbs. per square inch is supplied by a pump for working a hydraulic press, and a steam accumulator having a 6-inch ram and 30-inch stroke is connected to the pipe line to moderate the inequalities of pressure and the fluctuations of the supply. Calculate the size of the piston, if the effective steam pressure is 120 lbs. per square inch, and the loss due to friction of the packings of the accumulator is 6 per cent. Also determine the work which the accumulator can perform for one stroke of the piston.

10. Describe by aid of sketches the construction and mode of action of a low-lift centrifugal pump. Show that in a pump of this kind with no whirlpool chamber, the peripheral velocity V of the wheel is given by the formula $V_o = \sqrt{(2gH + u_o^2 \operatorname{cosec}^2 \phi)}$, where u_o is the radial outlet velocity, and ϕ is the angle of the blades at the periphery of the wheel. Show what effect the variation of the angle ϕ has upon the velocity of the wheel, and describe briefly the special advantages of pumps (*a*) with radial blades at outlet, (*b*) with re-curved blades.

11. Describe the construction and action of a Pitot tube, and show how you would use it to obtain the velocity of water at different points in the section of a water main. Sketch the probable form of the velocity curve for a main in which the mean velocity of water is, say, 5 feet per second (1) at a section in a straight length of pipe, (2) at a bend of moderate curvature, say, of 4 diameters.

12. Give a short account of the results of experiments on the dissipation of head due to bends, elbows and sudden changes of section in a pipe, and deduce an expression for the loss of head due to sudden enlargement in a pipe line. A 4-inch main conveying water at a velocity of 10 feet per second is suddenly enlarged to 6 inches in diameter. Calculate the loss of head due to the change of section.

THE INSTITUTION OF CIVIL ENGINEERS' EXAMINATION, OCTOBER, 1909.

ELECTION OF ASSOCIATE MEMBERS.

HYDRAULICS.

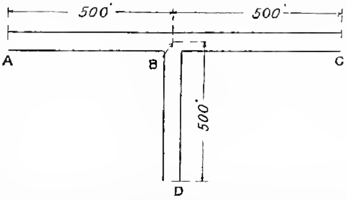
(*Not more than* EIGHT *questions to be attempted by any Candidate.*)

1. Twelve hundred gallons of water per minute are to be lifted a height of 150 feet through a pipe 1 foot diameter and 5,000 feet long. Assuming the head lost by friction in the pipe per foot length when the velocity is 1 foot per second is 0·0005 foot, and the mechanical efficiency of the pump 80 per cent., find the effective horse-power of the engine driving the pump.

2. The outlet circumference of a centrifugal pump has a diameter of 2 feet, and the inlet circumference is 1 foot 3 inches diameter. The radial velocity of flow at inlet is 5 feet and at outlet 8 feet per second. The wheel makes 400 revolutions per minute. The vanes are radial at exit. Determine the theoretical lift of the pump and the direction of the inlet edge of the vane.

3. Show that Bernoulli's theorem is consistent with the principle of the conservation of energy. One cubic foot of water per second flows along a pipe, which has a slope of 1 in 10, from A to B. At A the diameter is 3 inches and at B 6 inches, and the horizontal distance from A to B is 50 feet. The pressure head at A is 20 feet of water and at B 25 feet. Find, from this data, the head lost between A and B.

4. A turbine is driven by a head of 120 feet. The water used in a test of the turbine is measured by passing it through a right-angled V notch. The mean head over the apex of the notch during the trial is found to be 9 inches. The brake-horse-power of the turbine is found to be 13; find the efficiency of the turbine.



5. A horizontal pipe, A C in the figure, has a diameter of 6 inches and is 1,000 feet long. At point B a horizontal branch-pipe 3 inches diameter and 500 feet long is taken off A C. The pressures at A, C, and D are found to be 20 feet, 10 feet, and 7·5 feet of water respectively. Assuming the head lost by friction in a

length l of either of the pipes to be $\frac{0{\cdot}0006\, l\, v^2}{d}$, and neglecting other losses, show that the velocity in A B is about 3·16 feet per second, and find the pressure at B.

6. Water issues from a square-edged vertical circular orifice 2 inches diameter, made in the side of a vessel in which a constant pressure of 200 lbs. per square inch is maintained, and the head of water in the vessel above the centre of the orifice is 8 feet. A flat plate is placed near to the orifice, the plate being inclined at an angle of 60° to the horizontal plane. Find the discharge from the orifice and the horizontal pressure on the plate.

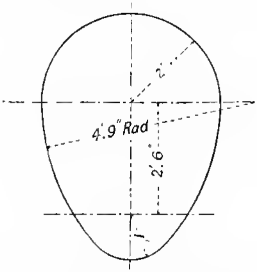
7. Describe briefly, with the aid of sketches, two methods of determining the discharge of a river 100 feet wide.

8. Make sketches of and explain the principle of action of one of the following:—(*a*) A compressed air pump showing the lift, and the depth of water in the well; (*b*) a hydraulic ram; (*c*) a hydraulic press in which the ram is returned automatically.

9. Make sketches of two arrangements for regulating the flow of water to pressure turbines and of one arrangement for regulating the flow to one form of impulse turbine. Discuss the effects of the arrangements in diminishing the efficiency of the turbines.

10. A hydraulic direct-acting lift has a ram 9 inches diameter. The pipe connecting the valve-box to the cylinder is $\frac{3}{4}$ inch diameter. The pressure in the valve-box is 600 lbs. per square inch. Neglecting all other losses than that due to sudden enlargement when the water enters the cylinder, and assuming the valve fully open at all loads, find the maximum load that can be lifted at a velocity of 1 foot per second. Find also the maximum velocity with which the lift could descend, assuming the exhaust full open.

11. A brick-lined sewer has a section shown in the figure, and a slope of 1 in 1,000. Determine the discharge of the sewer in cubic feet per second when the depth of flow is 2 feet. Explain the formula you use. (*Note.*—Areas, &c., may be found from a scale drawing.)



12. Give an account of any experiments with which you are familiar on the discharge of water over weirs. State the deductions to which the experiments would lead you in making a weir for the accurate gauging of water, with respect to the following points:—(*a*) The form of the sill of the weir, (*b*) the form of the approaching channel, (*c*) any precautions to be taken in connection with the down-stream channel.

THE INSTITUTION OF CIVIL ENGINEERS EXAMINATION, FEBRUARY, 1910.

ELECTION OF ASSOCIATE MEMBERS.

HYDRAULICS.

Not more than EIGHT *questions to be attempted by any Candidate.*

1. The discharges over a weir under different heads were as follows :—

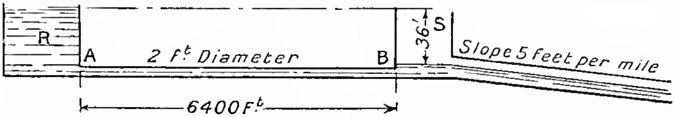
Head.	Discharge in Cubic Feet per Second.
0·5	11·5
1·0	33·3
1·5	61·5

On a certain day the following readings of the head over the weir were taken :—

Time.	Head.
2 o'clock	1·0
3 ,,	1·125
4 ,,	1·250

Plot a head-discharge curve for the weir, and also a time-discharge curve, and determine approximately the discharge over the weir between 2 o'clock and 4 o'clock.

2. A pipe A B, 2 feet in diameter, conveys water from a reservoir to a sump S. From S the water runs along a circular culvert having a slope of 5 feet per mile. When the difference of level of the water in the reservoir and the sump is 36 feet, the water in the aqueduct is required to be flowing steadily at a level coinciding with the centre. Find the diameter of the culvert. The value of C in the formula $v = C\sqrt{m\,i}$ may be taken as 100 for the pipe and culvert.



3. Two vessels are connected by a horizontal pipe 3 inches diameter and 20 feet in length. The surface of the water in one vessel is kept at a height of 10 feet above the centre of the pipe, and the pressure above the surface is 14·7 lbs. per square inch absolute. The surface of the water in the second vessel is kept at 3 feet above the centre of the pipe, and the pressure above the surface is 2 lbs. per square inch absolute. The pipe is not bell-mouthed at either end. Find the discharge per second into the second vessel. The head lost by friction is $h = \dfrac{0{\cdot}04\, l\, v^2}{2\, g\, d}$.

4. The tanks of the last question are each 12 feet long by 12 feet wide. Assuming that the water in the first vessel is at any instant 10 feet above

the centre of the pipe, and the surface of the water in the other is 3 feet above the centre, and that no water enters the first vessel while flow takes place into the second, and the pressure above the surfaces remains constant, find the time taken for the surfaces to come to the same level.

5. The plunger of a pump is 6 inches diameter, its stroke is 1 foot, and it makes 30 strokes per minute. The pump has an air vessel on the delivery. The mean pressure at the delivery valve of the pump is 700 lbs. per square inch. The pump delivers water to a crane having a ram 10 inches diameter through a pipe 1 inch diameter. The velocity ratio of the crane hook to the ram is 6. Neglecting all losses, except loss by sudden enlargement when the water enters the crane cylinder, find the maximum load the crane can lift. The mechanical efficiency of the crane is 85 per cent.

6. Sketch a Poncelet wheel, or some form of impulse turbine, and show clearly how you would determine the form of the vanes for maximum efficiency. Show that for a Poncelet wheel, neglecting friction, the most efficient velocity of the wheel rim is $v = \frac{1}{2}\cos\theta\sqrt{2gH}$. H = head. θ = the angle between the approaching water and the tangent to the wheel.

7. Make sketches showing the construction of either :—(*a*) Some form of high lift centrifugal pump, showing clearly the arrangement for converting velocity head into pressure head, or (*b*) a reciprocating pump suitable for supplying water to hydraulic cranes at a pressure of 750 lbs. per square inch, showing clearly the arrangement of the valves, and the packings for the plunger and the glands.

8. Describe briefly, and show by sketches, how you would determine the loss of head in any length of a pipe of constant diameter along which water is flowing, and hence from a number of experiments how you would deduce a formula to express the loss of head at any velocity in, say, clean cast-iron pipes. Quote the formulæ you would be likely to get for (*a*) clean cast-iron pipes ; (*b*) clean brass pipes.

9. Show that in a turbine the work done per lb. of water on the wheel is—

$$\frac{Vv}{g} - \frac{V_1 v_1}{g} = H - \frac{\mu^2}{2g}.$$

V = velocity of whirl at inlet ;
V_1 = ,, ,, ,, outlet ;
v = ,, ,, wheel at inlet ;
v_1 = ,, ,, ,, outlet ;
H = head under which turbine works ;
μ = velocity of the water as it leaves the turbine.

10. A jet delivers 100 cubic feet of water per second under a head of 100 feet, and strikes a plane inclined at 45° to the direction of the jet. Calculate the normal pressure on the plane.

11. The inner and outer diameters of an inward flow turbine wheel are 1 foot 6 inches and 3 feet respectively, and the velocity of the wheel periphery is 45 feet per second. The radial velocity of flow is 8 feet per second. The water leaves the guide blades in a direction making 20° with the tangent to the wheel, and leaves the wheel radially. Determine the angles of the wheel vanes at both circumferences, and the head under which the wheel is working.

12. Make sketches of the following :—A hydraulic crane cylinder in which the quantity of water supplied for a given lift can be diminished when the load is small, or a hydraulic engine in which the quantity of water used is regulated according to the work done.

City and Guilds of London Institute.

DEPARTMENT OF TECHNOLOGY.

TECHNOLOGICAL EXAMINATIONS,

1909.

For General Instructions, see Appendix A.

46.—MECHANICAL ENGINEERING.

Honours Grade (Written Examination).

The Candidate is not expected to answer more than *eight* of the following questions, which must be selected from *two* sections *only*.

Section A is given in Vol. II., "Strength of Materials," and Section B in the Author's *Text-Book on Steam and Steam Engines, including Turbines and Boilers.*

Section C (Hydraulics).

1. Prove that the discharge Q from a triangular notch is expressed by the formula—

$$Q = \frac{4}{15} c \sqrt{2g} B H^{\frac{3}{2}},$$

where H is the head of water, B is its surface breadth, and c is a constant derived from experiment. A right-angled notch is used to gauge the flow from a channel, and the average head is found to be 9·32 inches. Calculate the discharge per second if the coefficient of the weir is 0·617.
(C. & G., 1909, H., Sec. C.)

2. Describe the principle of action of the Venturi water meter, and obtain a formula for the discharge through it. A meter of this form in a 12-inch water main has a throat 5 inches in diameter and the difference of head between the upstream section and the throat is 21 feet. Calculate the discharge from the meter per second on the assumption that its coefficient is unity. (C. & G., 1909, H., Sec. C.)

3. Describe the principal features in a modern hydraulic power station for distributing water at a high pressure for use in operating lifts, cranes, hydraulic presses, and the like. (C. & G., 1909, H., Sec. C.)

4. The ram of a hydraulic press, using water at 2,000 lbs. per square inch, has a diameter of 1 foot and a stroke of 8 inches. It is operated by an intensifier, which latter is supplied with water from the town main at a pressure of 30 lbs. per square inch. Sketch to scale a suitable form of such an intensifier to give a full stroke of the ram for a single stroke of the intensifier and mark on it the principal dimensions, explaining how you have determined them. (C. & G., 1909, H., Sec. C.)

5. A centrifugal pump is required to lift 800 gallons of water per minute through a height of 20 feet. Briefly describe, by aid of sketches, the construction of a pump suitable for this head and delivery, and, assuming all the formulæ necessary, calculate the principal dimensions of the pump and mark them on your sketches. (C. & G., 1909, H., Sec. C.)

6. Assuming the loss of head h in a cast-iron main of length l and diameter d to be expressed by—

$$h = \frac{\cdot 0004\, v^{1\cdot 87}}{d^{1\cdot 4}}$$

in feet and second units, v being the velocity of the water, determine the loss of head due to frictional resistances in a cast-iron water main 2 feet in diameter and 2 miles in length if the mean velocity is 4 feet per second. (C. & G., 1909, H., Sec. C.)

7. Describe, by aid of sketches, the construction and mode of action of one form of multi-stage turbine pump, and explain what features of turbine design have been successfully introduced to obtain a very high lift. (C. & G., 1909, H., Sec. C.)

1910 MAY EXAMINATION ON SUBJECT VII. APPLIED MECHANICS.*

(b) MACHINES AND HYDRAULICS.

BY THE BOARD OF EDUCATION SECONDARY BRANCH,
SOUTH KENSINGTON, LONDON.

FOR GENERAL INSTRUCTIONS, SEE APPENDIX A.

STAGE 2.†

INSTRUCTIONS.

You must not attempt more than EIGHT *questions in all, including Nos.* 21 *and* 22*—that is to say, although* 21 *and* 22 *are not compulsory, you are not allowed to take more than six questions in addition to Nos.* 21 *and* 22.

4. The wheel of a centrifugal pump is delivering very little water, show in a curve how the pressure varies from inside the wheel to a point some distance outside the rim. Show this when there is a considerable delivery of water. Draw the shape of the vane of the wheel.
(B. of E., S. 2, Div. *b*, 1910.)

6. A particle of water is at a place A where the pressure is 0, its height above datum is 50 feet, its velocity is 5 feet per second, and it finds its way without friction to a place B, where the pressure is 0, and height above datum 30 feet; what is its velocity at B? (B. of E., S. 2, Div. *b*, 1910.)

8. A particle of air flows without friction. Its pressure p (in pounds per square foot) and its speed v (in feet per second) may both alter, but the sum

$$\frac{v^2}{2g}+\frac{p}{w}$$

remains constant, if w is the average weight of a cubic foot of air, g is 32·2. At a place A, p is 1·1 atmospheres and v is 0; a particle finds its way from A to B. At B the pressure is 1 atmosphere, what is the speed at B? Take w as ·075 lb. per cubic foot. (B. of E., S. 2, Div. *b*, 1910.)

11. The radial speed of the water in the wheel of a centrifugal pump is 6 feet per second; the vanes are directed backwards at an angle of 35° to the rim; what is the real velocity of the water relatively to the vanes? What is the component of this which is tangential to the rim? When the water has left the wheel, what is its velocity in the tangential direction if the rim of the wheel moves at 20 feet per second?
(B. of E., S. 2, Div. *b*, 1910.)

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[*] Students are referred to Vols. II. and III. for Questions under Div. (*a*) *Materials and Structures.*

[†] See Vol. I.—*Applied Mechanics*—for Questions 7, 10, 12, 15, 24, 27, 29, 31, and 32, and Vol. V.—*Theory of Machines*—for Questions 3, 5, 9, 13, 14, 21, 22, 28, 30, 33, and 34.

STAGE 3.

Read the instructions given under Stage 2.

23. What is meant by "Head" (*a*) in flowing water, (*b*) in flowing air? The following measurements have been made of i, the loss of head per unit length of a straight pipe through which water is flowing, v being the velocity of flow. Find the law connecting i and v:—

i.	v.	i.	v.
0·0086	0·239	0·1378	1·987
0·0172	0·451	0·1714	2·203
0·0258	0·690	0·3014	3·141
0·0345	0·914	0·4306	3·93
0·0430	1·109	0·8185	5·66
0·0516	1·349		

(B. of E., S. 3, Div. *b*, 1910.)

25. A 2-inch horizontal pipe through which water is flowing with a velocity of 12 feet per second suddenly enlarges to a diameter of $4\frac{1}{2}$ inches. Determine the loss of head due to the enlargement. Is your formula a mere empirical formula deduced from experiments, or is it a rational formula? If it is a rational formula, prove it. Criticise the proof.

(B. of E., S. 3, Div. *b*, 1910.)

26. An inward flow turbine, when running steadily, uses 40 cubic feet of water per second. The water enters the wheel with an absolute velocity of 45 feet per second, and its direction makes an angle of 23° with the direction of motion of the wheel rim; it leaves the wheel with an absolute velocity of 8 feet per second, and its direction makes an angle of 112° with the direction of motion of the wheel rim. The internal and external radii of the wheel are 1·6 and 2·8 feet respectively. What horse-power is the turbine receiving? State the principle on which you calculate.

(B. of E., S. 3, Div. *b*, 1910.)

APPENDIX D.

EXAMINATION TABLES.

USEFUL CONSTANTS.

1 Inch = 25·4 millimetres.

1 Gallon = ·1605 cubic foot = 10 lbs. of water at 62° F. ∴ 1 lb. = ·01605 cubic foot.

1 Knot = 6080 feet per hour. 1 Naut = 6080 feet.

Weight of 1 lb. in London = 445,000 dynes.

One pound avoirdupois = 7000 grains = 453·6 grammes.

1 Cubic foot of water weighs 62·3 lbs.

1 Cubic foot of air at 0° C. and 1 atmosphere, weighs ·0807 lb.

1 Cubic foot of Hydrogen at 0° C. and 1 atmosphere, weighs ·00557 lb.

1 Foot-pound = $1{\cdot}3562 \times 10^7$ ergs.

1 Horse-power-hour = 33000 × 60 foot-pounds.

1 Electrical unit = 1000 watt-hours.

Joule's Equivalent to suit Regnault's H, is $\left\{\begin{array}{l}\text{774 ft.-lbs. = 1 Fah. unit.}\\ \text{1393 ft.-lbs. = 1 Cent. ,,}\end{array}\right.$

1 Horse-power = 33000 foot-pounds per minute = 746 watts.

Volts × amperes = watts.

1 Atmosphere = 14·7 lb. per square inch = 2116 lbs. per square foot = 760 m.m. of mercury = 10^6 dynes per sq. cm. nearly.

A Column of water 2·3 feet high corresponds to a pressure of 1 lb. per square inch.

Absolute temp., $t = \theta° \text{ C.} + 273°{\cdot}7$.

Regnault's H = 606·5 + ·305 θ° C = 1082 + ·305 θ° F.

$$p\,u^{1{\cdot}0646} = 479$$

$$\log_{10} p = 6{\cdot}1007 - \frac{B}{t} - \frac{C}{t^2},$$

where $\log_{10} B = 3{\cdot}1812$, $\log_{10} C = 5{\cdot}0871$,

p is in pounds per square inch, t is absolute temperature Centigrade, u is the volume in cubic feet per pound of steam.

$$\pi = 3{\cdot}1416 = \frac{22}{7} = \frac{355}{113} = 10(\sqrt{3} - \sqrt{2}).$$

One radian = 57·3 degrees.

To convert common into Napierian logarithms, multiply by 2·3026.

The base of the Napierian logarithm is $e = 2{\cdot}7183$.

The value of g at London = 32·182 feet per second per second.

TABLE OF LOGARITHMS.

	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
10	0000	0043	0086	0128	0170	0212	0253	0294	0334	0374	4	8	12	17	21	25	29	33	37
11	0414	0453	0492	0531	0569	0607	0645	0682	0719	0755	4	8	11	15	19	23	26	30	34
12	0792	0828	0864	0899	0934	0969	1004	1038	1072	1106	3	7	10	14	17	21	24	28	31
13	1139	1173	1206	1239	1271	1303	1335	1367	1399	1430	3	6	10	13	16	19	23	26	29
14	1461	1492	1523	1553	1584	1614	1644	1673	1703	1732	3	6	9	12	15	18	21	24	27
15	1761	1790	1818	1847	1875	1903	1931	1959	1987	2014	3	6	8	11	14	17	20	22	25
16	2041	2068	2095	2122	2148	2175	2201	2227	2253	2279	3	5	8	11	13	16	18	21	24
17	2304	2330	2355	2380	2405	2430	2455	2480	2504	2529	2	5	7	10	12	15	17	20	22
18	2553	2577	2601	2625	2648	2672	2695	2718	2742	2765	2	5	7	9	12	14	16	19	21
19	2788	2810	2833	2856	2878	2900	2923	2945	2967	2989	2	4	7	9	11	13	16	18	20
20	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201	2	4	6	8	11	13	15	17	19
21	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404	2	4	6	8	10	12	14	16	18
22	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598	2	4	6	8	10	12	14	15	17
23	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784	2	4	6	7	9	11	13	15	17
24	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962	2	4	5	7	9	11	12	14	16
25	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133	2	3	5	7	0	10	12	14	15
26	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298	2	3	5	7	8	10	11	13	15
27	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456	2	3	5	6	8	9	11	13	14
28	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609	2	3	5	6	8	9	11	12	14
29	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757	1	3	4	6	7	9	10	12	13
30	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900	1	3	4	6	7	9	10	11	13
31	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038	1	3	4	6	7	8	10	11	12
32	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172	1	3	4	5	7	8	9	11	12
33	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302	1	3	4	5	6	8	9	10	12
34	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428	1	3	4	5	6	8	9	10	11
35	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551	1	2	4	5	6	7	9	10	11
36	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670	1	2	4	5	6	7	8	10	11
37	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786	1	2	3	5	6	7	8	9	10
38	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899	1	2	3	5	6	7	8	9	10
39	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010	1	2	3	4	5	7	8	9	10
40	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117	1	2	3	4	5	6	8	9	10
41	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222	1	2	3	4	5	6	7	8	9
42	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325	1	2	3	4	5	6	7	8	9
43	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425	1	2	3	4	5	6	7	8	9
44	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522	1	2	3	4	5	6	7	8	9
45	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618	1	2	3	4	5	6	7	8	9
46	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712	1	2	3	4	5	6	7	7	8
47	6721	6730	6739	6749	6758	6767	6776	6785	6704	6803	1	2	3	4	5	5	6	7	8
48	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893	1	2	3	4	4	5	6	7	8
49	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981	1	2	3	4	4	5	6	7	8
50	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067	1	2	3	3	4	5	6	7	8
51	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152	1	2	3	3	4	5	6	7	8
52	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235	1	2	2	3	4	5	6	7	7
53	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316	1	2	2	3	4	5	6	6	7
54	7324	7332	7340	7348	7356	7364	7372	7380	7388	7396	1	2	2	3	4	5	6	6	7

TABLE OF LOGARITHMS.—*Continued.*

	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
65	7404	7412	7419	7427	7436	7443	7451	7459	7466	7474	1	2	2	3	4	5	5	6	7
56	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551	1	2	2	3	4	5	6	6	7
57	7559	7566	7574	7582	7689	7597	7604	7612	7619	7627	1	2	2	3	4	5	5	6	7
58	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701	1	1	2	3	4	4	5	6	7
59	7709	7716	7723	7731	7738	7746	7752	7760	7767	7774	1	1	2	3	4	4	5	6	7
60	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846	1	1	2	3	4	4	5	6	6
61	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917	1	1	2	3	4	4	5	6	6
62	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987	1	1	2	3	3	4	5	6	6
63	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055	1	1	2	3	3	4	5	5	6
64	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122	1	1	2	3	3	4	5	5	6
65	8129	8136	8142	8149	8156	8162	8169	8176	8182	8189	1	1	2	3	3	4	5	5	6
66	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254	1	1	2	3	3	4	5	5	6
67	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319	1	1	2	3	3	4	5	5	6
68	8325	8331	8338	8344	8351	8357	8363	8370	8376	8382	1	1	2	3	3	4	4	5	6
69	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445	1	1	2	2	3	4	4	5	6
70	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506	1	1	2	2	3	4	4	5	6
71	8513	8519	8525	8531	8537	8543	8549	8556	8661	8567	1	1	2	2	3	4	4	5	5
72	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627	1	1	2	2	3	4	4	5	5
73	8633	8639	8645	8651	8657	8663	8669	8676	8681	8686	1	1	2	2	3	4	4	5	6
74	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745	1	1	2	2	3	4	4	5	5
75	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802	1	1	2	2	3	3	4	5	5
76	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859	1	1	2	2	3	3	4	5	5
77	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915	1	1	2	2	3	3	4	4	5
78	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971	1	1	2	2	3	3	4	4	5
79	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025	1	1	2	2	3	3	4	4	5
80	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079	1	1	2	2	3	3	4	4	5
81	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133	1	1	2	2	3	3	4	4	6
82	9138	9143	9149	9154	9159	9165	9170	9175	9180	9186	1	1	2	2	3	3	4	4	5
83	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238	1	1	2	2	3	3	4	4	5
84	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289	1	1	2	2	3	3	4	4	5
85	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340	1	1	2	2	3	3	4	4	5
86	9346	9350	9355	9360	9365	9370	9375	9380	9385	9390	1	1	2	2	3	3	4	4	5
87	9395	9400	9405	9410	9415	9420	9425	9430	9436	9440	0	1	1	2	2	3	3	4	4
88	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489	0	1	1	2	2	3	3	4	4
89	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538	0	1	1	2	2	3	3	4	4
90	9542	9547	9552	9557	9562	9566	9571	9576	9581	9586	0	1	1	2	2	3	3	4	4
91	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633	0	1	1	2	2	3	3	4	4
92	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680	0	1	1	2	2	3	3	4	4
93	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727	0	1	1	2	2	3	3	4	4
94	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773	0	1	1	2	2	3	3	4	4
95	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818	0	1	1	2	2	3	3	4	4
96	9823	9827	9832	9836	9841	9846	9850	9854	9859	9863	0	1	1	2	2	3	3	4	4
97	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908	0	1	1	2	2	3	3	4	4
98	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952	0	1	1	2	2	3	3	4	4
99	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996	0	1	1	2	2	3	3	3	4

TABLE OF ANTILOGARITHMS.

	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
·00	1000	1002	1005	1007	1009	1012	1014	1016	1019	1021	0	0	1	1	1	1	2	2	2
·01	1023	1026	1028	1030	1033	1035	1038	1040	1042	1045	0	0	1	1	1	1	2	2	2
·02	1047	1050	1052	1054	1057	1059	1062	1064	1067	1069	0	0	1	1	1	1	2	2	2
·03	1072	1074	1076	1079	1081	1084	1086	1089	1091	1094	0	0	1	1	1	1	2	2	2
·04	1096	1099	1102	1104	1107	1109	1112	1114	1117	1119	0	1	1	1	1	2	2	2	2
·05	1122	1125	1127	1130	1132	1135	1138	1140	1143	1146	0	1	1	1	1	2	2	2	2
·06	1148	1151	1153	1156	1159	1161	1164	1167	1169	1172	0	1	1	1	1	2	2	2	2
·07	1175	1178	1180	1183	1186	1189	1191	1194	1197	1199	0	1	1	1	1	2	2	2	2
·08	1202	1205	1208	1211	1213	1216	1219	1222	1225	1227	0	1	1	1	1	2	2	2	3
·09	1230	1233	1236	1239	1242	1245	1247	1250	1253	1256	0	1	1	1	1	2	2	2	3
·10	1259	1262	1265	1268	1271	1274	1276	1279	1282	1285	0	1	1	1	1	2	2	2	3
·11	1288	1291	1294	1297	1300	1303	1306	1309	1312	1315	0	1	1	1	2	2	2	2	3
·12	1318	1321	1324	1327	1330	1334	1337	1340	1343	1346	0	1	1	1	2	2	2	2	3
·13	1349	1352	1355	1358	1361	1365	1368	1371	1374	1377	0	1	1	1	2	2	2	3	3
·14	1380	1384	1387	1390	1393	1396	1400	1403	1406	1409	0	1	1	1	2	2	2	3	3
·15	1413	1416	1419	1422	1426	1429	1432	1435	1439	1442	0	1	1	1	2	2	2	3	3
·16	1445	1449	1452	1455	1459	1462	1466	1469	1472	1476	0	1	1	1	2	2	2	3	3
·17	1479	1483	1486	1489	1493	1496	1500	1503	1507	1510	0	1	1	1	2	2	2	3	3
·18	1514	1517	1521	1524	1528	1531	1535	1538	1542	1545	0	1	1	1	2	2	2	3	3
·19	1549	1552	1556	1560	1563	1567	1570	1574	1578	1581	0	1	1	1	2	2	3	3	3
·20	1585	1589	1592	1596	1600	1603	1607	1611	1614	1618	0	1	1	1	2	2	3	3	3
·21	1622	1626	1629	1633	1637	1641	1644	1648	1652	1656	0	1	1	2	2	2	3	3	3
·22	1660	1663	1667	1671	1675	1679	1683	1687	1690	1694	0	1	1	2	2	2	3	3	3
·23	1698	1702	1706	1710	1714	1718	1722	1726	1730	1734	0	1	1	2	2	2	3	3	4
·24	1738	1742	1746	1750	1754	1758	1762	1766	1770	1774	0	1	1	2	2	2	3	3	4
·25	1778	1782	1786	1791	1795	1799	1803	1807	1811	1816	0	1	1	2	2	2	3	3	4
·26	1820	1824	1828	1832	1837	1841	1845	1849	1854	1858	0	1	1	2	2	3	3	3	4
·27	1862	1866	1871	1875	1879	1884	1888	1892	1897	1901	0	1	1	2	2	3	3	3	4
·28	1905	1910	1914	1919	1923	1928	1932	1936	1941	1945	0	1	1	2	2	3	3	4	4
·29	1950	1954	1959	1963	1968	1972	1977	1982	1986	1991	0	1	1	2	2	3	3	4	4
·30	1995	2000	2004	2009	2014	2018	2023	2028	2032	2037	0	1	1	2	2	3	3	4	4
·31	2042	2046	2051	2056	2061	2065	2070	2075	2080	2084	0	1	1	2	2	3	3	4	4
·32	2089	2094	2099	2104	2109	2113	2118	2123	2128	2133	0	1	1	2	2	3	3	4	4
·33	2138	2143	2148	2153	2158	2163	2168	2173	2178	2183	0	1	1	2	2	3	3	4	4
·34	2188	2193	2198	2203	2208	2213	2218	2223	2228	2234	1	1	2	2	3	3	4	4	5
·35	2239	2244	2249	2254	2259	2265	2270	2275	2280	2286	1	1	2	2	3	3	4	4	5
·36	2291	2296	2301	2307	2312	2317	2323	2328	2333	2339	1	1	2	2	3	3	4	4	5
·37	2344	2350	2355	2360	2366	2371	2377	2382	2388	2393	1	1	2	2	3	3	4	4	5
·38	2399	2404	2410	2415	2421	2427	2432	2438	2443	2449	1	1	2	2	3	3	4	4	5
·39	2455	2460	2466	2472	2477	2483	2489	2495	2500	2506	1	1	2	2	3	3	4	5	5
·40	2512	2518	2523	2529	2535	2541	2547	2553	2559	2564	1	1	2	2	3	4	4	5	5
·41	2570	2576	2582	2588	2594	2600	2606	2612	2618	2624	1	1	2	2	3	4	4	5	5
·42	2630	2636	2642	2649	2655	2661	2667	2673	2679	2685	1	1	2	2	3	4	4	5	6
·43	2692	2698	2704	2710	2716	2723	2729	2735	2742	2748	1	1	2	3	3	4	4	5	6
·44	2754	2761	2767	2773	2780	2786	2793	2799	2805	2812	1	1	2	3	3	4	4	5	6
·45	2818	2825	2831	2838	2844	2851	2858	2864	2871	2877	1	1	2	3	3	4	5	5	6
·46	2884	2891	2897	2904	2911	2917	2924	2931	2938	2944	1	1	2	3	3	4	5	5	6
·47	2951	2958	2965	2972	2979	2985	2992	2999	3006	3013	1	1	2	3	3	4	5	5	6
·48	3020	3027	3034	3041	3048	3055	3062	3069	3076	3083	1	1	2	3	4	4	5	6	6
·49	3090	3097	3105	3112	3119	3126	3133	3141	3148	3155	1	1	2	3	4	4	5	6	6

TABLE OF ANTILOGARITHMS.—*Continued.*

	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
·50	3162	3170	3177	3184	3192	3199	3206	3214	3221	3228	1	1	2	3	4	4	5	6	7
·51	3236	3243	3251	3258	3266	3273	3281	3289	3296	3304	1	2	2	3	4	5	5	6	7
·52	3311	3319	3327	3334	3342	3350	3357	3365	3373	3381	1	2	2	3	4	5	5	6	7
·53	3388	3396	3404	3412	3420	3428	3436	3443	3451	3459	1	2	2	3	4	5	6	6	7
·54	3467	3475	3483	3491	3499	3508	3516	3524	3532	3540	1	2	2	3	4	5	6	6	7
·55	3548	3556	3565	3573	3581	3589	3597	3606	3614	3622	1	2	2	3	4	5	6	7	7
·56	3631	3639	3648	3656	3664	3673	3681	3690	3698	3707	1	2	3	3	4	5	6	7	8
·57	3715	3724	3733	3741	3750	3758	3767	3776	3784	3793	1	2	3	3	4	5	6	7	8
·58	3802	3811	3819	3828	3837	3846	3855	3864	3873	3882	1	2	3	4	4	5	6	7	8
·59	3890	3899	3908	3917	3926	3936	3945	3954	3963	3972	1	2	3	4	5	5	6	7	8
·60	3981	3990	3999	4009	4018	4027	4036	4046	4055	4064	1	2	3	4	5	6	6	7	8
·61	4074	4083	4093	4102	4111	4121	4130	4140	4150	4159	1	2	3	4	5	6	7	8	9
·62	4169	4178	4188	4198	4207	4217	4227	4236	4246	4256	1	2	3	4	5	6	7	8	9
·63	4266	4276	4285	4295	4305	4315	4325	4336	4345	4355	1	2	3	4	5	6	7	8	9
·64	4365	4375	4385	4395	4406	4416	4426	4436	4446	4457	1	2	3	4	5	6	7	8	9
·65	4467	4477	4487	4498	4508	4519	4529	4539	4550	4560	1	2	3	4	5	6	7	8	9
·66	4571	4581	4592	4603	4613	4624	4634	4645	4656	4567	1	2	3	4	5	6	7	9	10
·67	4677	4688	4699	4710	4721	4732	4742	4753	4764	4775	1	2	3	4	5	7	8	9	10
·68	4786	4797	4808	4819	4831	4842	4853	4864	4875	4887	1	2	3	4	6	7	8	9	10
·69	4898	4909	4920	4932	4943	4955	4966	4977	4989	5000	1	2	3	5	6	7	8	9	10
·70	5012	5023	5035	5047	5058	5070	5082	5093	5105	5117	1	2	4	5	6	7	8	9	11
·71	5129	5140	5152	5164	5176	5188	5200	5212	5224	5236	1	2	4	5	6	7	8	10	11
·72	5248	5260	5272	5284	5297	5309	5321	5333	5346	5358	1	2	4	5	6	7	9	10	11
·73	5370	5383	5395	5408	5420	5433	5445	5458	5470	5483	1	3	4	5	6	8	9	10	11
·74	5495	5508	5521	5534	5546	5559	5572	5585	5598	5610	1	3	4	5	6	8	9	10	12
·75	5623	5636	5649	5662	5675	5689	5702	5715	5728	5741	1	3	4	5	7	8	9	10	12
·76	5754	5768	5781	5794	5808	5821	5834	5848	5861	5875	1	3	4	5	7	8	9	11	12
·77	5888	5902	5916	5929	5943	5957	5970	5984	5998	6012	1	3	4	5	7	8	10	11	12
·78	6026	6039	6053	6067	6081	6095	6109	6124	6138	6152	1	3	4	6	7	8	10	11	13
·79	6166	6180	6194	6209	6223	6237	6252	6266	6281	6295	1	3	4	6	7	9	10	11	13
·80	6310	6324	6339	6353	6368	6383	6397	6412	6427	6442	1	3	4	6	7	9	10	12	13
·81	6457	6471	6486	6501	6516	6531	6546	6561	6577	6592	2	3	5	6	8	9	11	12	14
·82	6607	6622	6637	6653	6668	6683	6699	6714	6730	6745	2	3	5	6	8	9	11	12	14
·83	6761	6776	6792	6808	6823	6839	6855	6871	6887	6902	2	3	5	6	8	9	11	13	14
·84	6918	6934	6950	6966	6982	6998	7015	7031	7047	7063	2	3	5	6	8	10	11	13	15
·85	7079	7096	7112	7129	7145	7161	7178	7194	7211	7228	2	3	5	7	8	10	12	13	15
·86	7244	7261	7278	7295	7311	7328	7345	7362	7379	7396	2	3	5	7	8	10	12	13	15
·87	7413	7430	7447	7464	7482	7499	7516	7534	7551	7568	2	3	5	7	9	10	12	14	16
·88	7586	7603	7621	7638	7656	7674	7691	7709	7727	7745	2	4	5	7	9	11	12	14	16
·89	7762	7780	7798	7816	7834	7852	7870	7889	7907	7925	2	4	5	7	9	11	13	14	16
·90	7943	7962	7980	7998	8017	8035	8054	8072	8891	8110	2	4	6	7	9	11	13	15	17
·91	8128	8147	8166	8185	8204	8222	8241	8260	8279	8299	2	4	6	8	9	11	13	15	17
·92	8318	8337	8356	8375	8395	8414	8433	8453	8472	8492	2	4	6	8	10	12	14	15	17
·93	8511	8531	8551	8570	8590	8610	8630	8650	8670	8690	2	4	6	8	10	12	14	16	18
·94	8710	8730	8750	8770	8790	8810	8831	8851	8872	8892	2	4	6	8	10	12	14	16	18
·95	8913	8933	8954	8974	8995	9016	9036	9057	9078	9099	2	4	6	8	10	12	15	17	19
·96	9120	9141	9162	9183	9204	9226	9247	9268	9290	9311	2	4	6	8	11	13	15	17	19
·97	9333	9354	9376	9397	9419	9441	9462	9484	9506	9528	2	4	7	9	11	13	15	17	20
·98	9550	9572	9594	9616	9638	9661	9683	9705	9727	9750	2	4	7	9	11	13	16	18	20
·99	9772	9795	9817	9840	9863	9886	9908	9931	9954	9977	2	5	7	9	11	14	16	18	20

TABLE OF FUNCTIONS OF ANGLES.

Angle.		Chord.	Sine.	Tangent.	Co-tangent.	Cosine.			
Degrees.	Radians.								
0°	0	000	0	0	∞	1	1·414	1·5708	90°
1	·0175	·017	·0175	·0175	57·2900	·9908	1·402	1·5533	89
2	·0349	·035	·0349	·0349	28·6363	·9994	1·389	1·5359	88
3	·0524	·052	·0523	·0524	19·0811	·9986	1·377	1·5184	87
4	·0698	·070	·0698	·0699	14·3007	·9976	1·364	1·5010	86
5	·0873	·087	·0872	·0875	11·4301	·9962	1·351	1·4835	85
6	·1047	·105	·1045	·1051	9·5144	·9945	1·338	1·4661	84
7	·1222	·122	·1219	·1228	8·1443	·9925	1·325	1·4486	83
8	·1396	·140	·1392	·1405	7·1154	·9903	1·312	1·4312	82
9	·1571	·157	·1564	·1584	6·3138	·9877	1·299	1·4137	81
10	·1745	·174	·1736	·1768	5·6713	·9848	1·286	1·3963	80
11	·1929	·192	·1908	·1944	5·1446	·9816	1·272	1·3788	79
12	·2094	·209	·2079	·2126	4·7046	·9781	1·259	1·3614	78
13	·2269	·226	·2250	·2309	4·3315	·9744	1·245	1·3439	77
14	·2443	·244	·2419	·2493	4·0108	·9703	1·231	1·3265	76
15	·2618	·261	·2588	·2679	3·7321	·9659	1·218	1·3090	75
16	·2793	·278	·2756	·2867	3·4874	·9613	1·204	1·2915	74
17	·2967	·296	·2924	·3057	3·2709	·9563	1·190	1·2741	73
18	·3142	·313	·3090	·3249	3·0777	·9511	1·176	1·2566	72
19	·3316	·330	·3256	·3443	2·9042	·9455	1·161	1·2392	71
20	·3491	·347	·3420	·3640	2·7475	·9397	1·147	1·2217	70
21	·3665	·364	·3584	·3839	2·6051	·9336	1·133	1·2043	69
22	·3840	·382	·3746	·4040	2·4751	·9272	1·118	1·1868	68
23	·4014	·399	·3907	·4245	2·3559	·9205	1·104	1·1694	67
24	·4189	·416	·4067	·4452	2·2460	·9135	1·089	1·1519	66
25	·4363	·433	·4226	·4663	2·1445	·9063	1·075	1·1345	65
26	·4538	·450	·4384	·4877	2·0503	·8988	1·060	1·1170	64
27	·4712	·467	·4540	·5095	1·9626	·8910	1·045	1·0006	63
28	·4887	·484	·4695	·5317	1·8807	·8829	1·030	1·0821	62
29	·5061	·501	·4848	·5543	1·8040	·8746	1·015	1·0647	61
30	·5236	·518	·5000	·5774	1·7321	·8660	1·000	1·0472	60
31	·5411	·534	·6150	·6009	1·6643	·8572	·985	1·0297	59
32	·5585	·551	·5299	·6249	1·6003	·8480	·970	1·0123	68
33	·5760	·568	·5446	·6494	1·5399	·8387	·954	·9948	57
34	·5934	·585	·5592	·6745	1·4826	·8290	·939	·9774	56
35	·6109	·601	·5736	·7002	1·4281	·8192	·923	·9599	55
36	·6283	·618	·5878	·7265	1·3764	·8090	·908	·9425	54
37	·6458	·635	·6018	·7536	1·3270	·7986	·892	·9250	53
38	·6632	·651	·6157	·7813	1·2799	·7880	·877	·9076	52
39	·6807	·668	·6293	·8098	1·2349	·7771	·861	·8901	51
40	·6981	·684	·6428	·8391	1·1918	·7660	·845	·8727	50
41	·7156	·700	·6561	·8693	1·1504	·7547	·829	·8552	49
42	·7330	·717	·6691	·9004	1·1106	·7431	·813	·8378	48
43	·7505	·733	·6820	·9325	1·0724	·7314	·797	·8203	47
44	·7679	·749	·6947	·9657	1·0356	·7193	·781	·8029	46
45°	·7854	·765	·7071	1·0000	1·0000	·7071	·765	·7854	45°
								Radians.	Degrees.
			Cosine.	Co-tangent.	Tangent.	Sine.	Chord.	Angle.	

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